POWER

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The Loose Pulley

AT AN exhibition of power-plant apparatus in connection with an engineers' convention recently, a prominent operating engineer was examining a recording device. "It is a good thing," he said; "they ought to be in general use, but lots of such things that I find around are on the loose pulley."

"How is that?" I asked. "On the loose pulley?"

"Shelved, sidetracked, out of use," he explained. "Some fellow gets a momentary fit of ambition; reads some circular or article or hears a lecture or gets set up by some smart salesman who convinces him that if he gets an indicator or recorder of this or that he will save tons of coal, thousands of dollars, and put himself in line to be a consulting engineer or a superintendent of power plants. And he would, if he used such things intelligently. But there has got to be brains in the outfit, and the machine has not got them. If the engineer cannot furnish them, nothing happens. He plays with it for a week or two, like a kid with a new toy, and then he gets tired of taking diagrams and records that don't mean anything to him and puts it on the 'loose pulley.' And when anybody asks him how he came out with it, he is likely to belittle the machine rather than to confess that he could not realize on its possibilities."

There is a world of truth in what he said. The efficiency of the average plant could be improved largely by an intelligent study of the results obtained in the different steps of converting and applying the energy of fuel. The necessary facts for such an analysis can be had only by the use of such apparatus. But one cannot make a poet out of a peasant by presenting him with a gold fountain pen. The need is for a combination of these instruments with the men who know how to use them. Where there are such combinations the instruments are earning big returns upon their cost. There are men who know how to use them who have not got them, and they and their plants would profit if their owners could be shown what they could accomplish with them.

And for those which are on the "loose pulley," how many unfulfilled dreams and unrealized good intentions have gone the same way? A little thought and study, consultation with the makers or with other engineers, reawakened interest, would realize the dream that filled the engineer's mind when he recommended their purchase, make good his representations to those who spent their money for them, improve the efficiency of his plant, add to his reputation as an engineer and help to conserve the fuel resources of the nation.

F. R. Low

Important Features in the Design of High-Head Reaction Turbines

A 30,000-Hp. Unit for Big Creek, No. 8 Development of the Southern California Edison Company—Leakage at Runner Seals—Leakage Around Guide Vanes—Draft Tube, Casing and Relief-Valve Design—Lubricating of Internal Parts—Governor Control

By F. H. ROGERS
Hydraulic Engineer, William Cramp and Sons Ship and Engine Building Company

APPLICATION of the reaction, or Francis type, hydraulic turbine is usually taken as for heads up to about 600 ft., although in 1911 two turbines were built for the Noriega plant of the Michoacan Power Co., in Mexico., for operation under a head of 670 ft., and there have recently been installed in the Kern River plant of the Southern California Edison Co., two reaction turbines designed for a head of 890 feet. There are many difficult problems to be solved in the successful application of reaction turbines to these high heads, and it is the writer's purpose to point out these problems and the manner in which they have been handled in the design and construction of a unit just completed by the I. P. Morris Department of the William Cramp and Sons Ship and Engine Building Co., for the Big Creek No. 8 Development of the Southern California Edison Co. This plant is one of a series of developments of the Edison company, located at the junction of Big Creek and the San Joaquin River in California. The turbine just completed is the first unit to be installed in this plant, but it is contemplated that the final capacity of the station will be about 120,000 horsepower.

The turbine is designed to develop 30,000 hp. under a 680-ft. head at a speed of 428 r.p.m., and is of the vertical-shaft, single-runner type. It is directly connected to a 22,500-kw. 14-pole 50-cycle 11,000-volt generator built by the General Electric Co. Fig. 1 shows a sectional elevation through the turbine and draft tube and outside elevations of the generator and relief valve. The water is led to the turbine through a penstock 2,800 ft. long, having a diameter of 7 ft. at the top and 6 ft. at the bottom, at which point is located a Johnson valve having an intake diameter of 6 ft. and a discharge diameter of 4 ft. 6 in. This valve was designed by the Larner-Johnson Valve and Engineering Co. and was built by the manufacturers of the turbine.

The Johnson valve is connected to the turbine casing through a cast-steel pipe containing a branch connection for the relief valve. A casing of the volute type, made of cast steel, is used. The movable guide vanes are of cast steel and the runner and seal rings are of bronze. The upper portion of the draft tube is of cast iron and the lower portion is formed in the concrete substructure of the power house. An oil-pressure type governor is belt-driven from the turbine shaft and is connected by piping to two operating cylinders which are directly connected by reach rods to the operating ring mounted on the turbine-head cover.

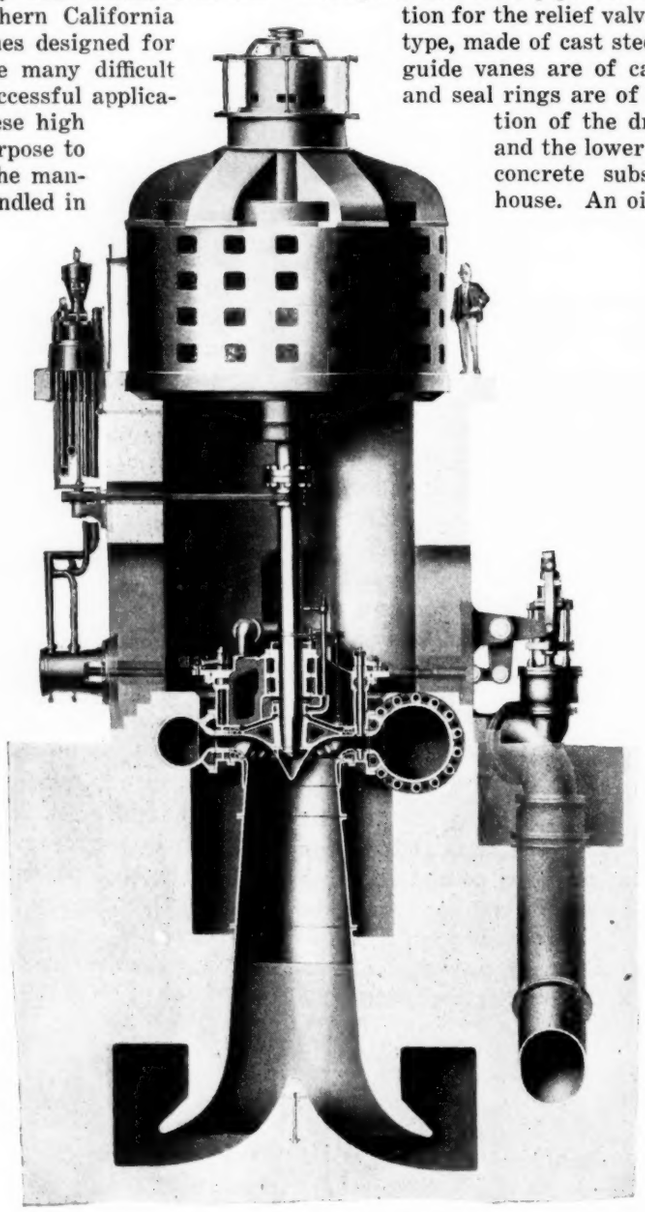


FIG. 1. SECTIONAL VIEW THROUGH TURBINE AND DRAFT TUBE AND OUTSIDE ELEVATION OF GENERATOR AND RELIEF VALVE

The particular problems to be studied will be discussed in the following order: (1) Leakage at the runner seals; (2) leakage around the guide vanes; (3) draft-tube design as affecting vibration; (4) casing design; (5) relief-valve design and pressure rise; (6) lubrication of internal parts; (7) governor control.

Leakage at the runner seals is an important factor in the design of high-head units. This leakage water escapes between the periphery of the runner and the stationary parts and hence performs no useful work. The amount of this leakage depends on the pressure existing at the intake to the runner and on the area between the

runner seals and the stationary parts. The diameter of this runner is 78.5 in., and the effective head under full-load operation at this point is about 349 ft. If for this runner the usual type of single straight seal were used, the leakage at the upper and lower seals would amount to approximately 12 cu.ft. per sec. if a clearance of $\frac{1}{32}$ in. were used. This leakage is 2.6 per cent of the full-load quantity required by the unit, and it is thus seen that even when the turbine is first installed this is a large item. The leakage can be considerably reduced by the use of the Moody labyrinth seal device illustrated in Fig.

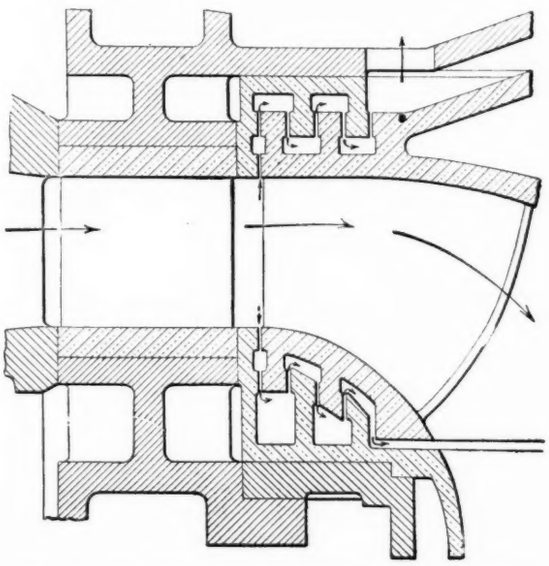


FIG. 2. SECTION THROUGH RUNNER SHOWING LABYRINTH SEALS

2, from which it will be noted that the leakage water must pass through a series of six seals, at each of which the velocity head is destroyed owing to changes in section and direction of flow. Hence, the effective head causing the leakage is only one-sixth of the head for a single seal. The use of this device on this particular runner will therefore reduce the leakage to about 3.9 cu.ft. per sec., which is about 0.9 per cent of the full-load quantity. It is thus seen that the labyrinth seals reduce the leakage to about one-third of that which would occur with the single seal.

This loss becomes of increasing importance as the operation of the unit is continued, since the wear at the seals is directly proportional to the velocity of flow through the seals, and as this velocity is about three times as great for the single seal as compared to the labyrinth seal, the wear at the former would be about three times as rapid as for the latter type. As an example, if after continuous operation for a period of a year, the original clearance should double for the single-seal design, the leakage would become about 24 cu.ft. per sec. For the labyrinth seal the increase in the clearance would be about $33\frac{1}{3}$ per cent, so that the total leakage would amount to about 5.2 cu.ft. per sec., and hence at the end of this period the leakage through the labyrinth seal would be only 22 per cent of the leakage occurring through the single seal.

Fig. 3 shows a view of the runner taken from the discharge side and indicates clearly the labyrinth seals on this side of the runner. These seals are machined out of the solid metal of the runner. The practice of attaching separate seals to the runner is in the writer's opinion, a dangerous one for two reasons: First, the peripheral speed causes a stress in the seal ring, tending to produce distortion, and secondly, any rubbing between the moving and the stationary seals causes heating, resulting in an expansion of the runner seals. Either of these conditions would probably result in the failure of the seal ring and serious damage to the runner or other internal parts. Even from the viewpoint of renewal there is little necessity for separate seal rings, as the purpose of the labyrinth seal is to reduce leakage and hence wear to a minimum.

Leakage Around Guide Vane

In the usual design of guide vane a constant flow occurs between the top and bottom of the vane and the distributor plates, and as this water is not properly guided by the vane, it strikes the runner vanes at an incorrect angle, causing considerable danger of cutting and pitting. For the same reason, when the guide vanes are closed, this leakage continues and will cause not only cutting on the runner vanes, but also wear on the distributor plates and the top and bottom of the guide vanes. It has been found in the case of high-head turbines that this leakage with the guide vanes closed, is often sufficient to keep the turbine runner rotating at considerable speed, so that it is necessary to close the penstock valve in order to bring the

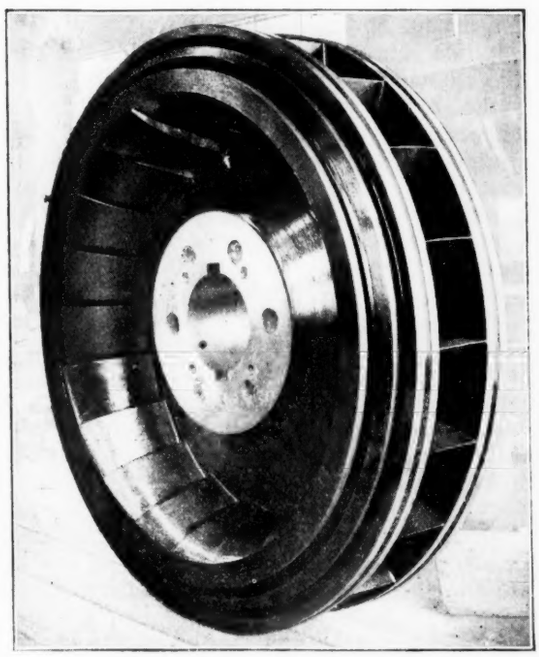


FIG. 3. VIEW OF RUNNER FROM DISCHARGE END

unit to rest. The guide vanes furnished for the Southern California Edison unit are of a special type, known as the Overn disk vane. A view of one of these vanes taken in the shop is shown in Fig. 4, from which it will be seen that two heavy disks are cast at the juncture between the vane and the upper and lower stems. When installed in the turbine, the faces of these disks are flush with the distributor plates, so that without disturbing the normal flow, they prevent any flow above or below the guide vane proper except for that small

portion of the vane which extends beyond the disk. By this means practically all the water is guided to the runner vanes at the proper angle and the danger of cutting the runner vanes, the guide vanes or the distributor plates is greatly reduced. For the same reason, when the vanes are in the closed position, there is very little leakage. This design further permits of a very much stronger bond between the guide vane and the steam, as the large diameter of the disk permits

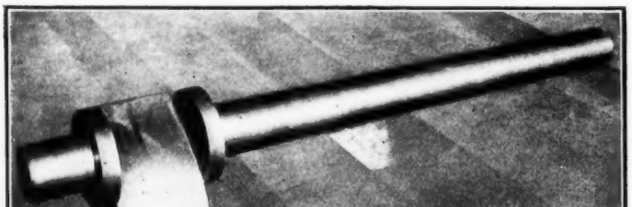


FIG. 4. GUIDE VANE OF THE OVERN DISK TYPE

ample area between the disk and the guide vane, which is very difficult to obtain in the usual design of vanes commonly used.

Draft-Tube Design

With this unit the draft tube provided is of the Moody spreading type, and by referring to Fig. 1, it will be seen that this design of tube meets the following important requirements: (1) The axis is a straight line coincident with the axis of the turbine; (2) the surrounding walls are surfaces of revolution about this axis; (3) the areas normal to the axial components are gradually enlarged and the tube spreads out symmetrically from the top to the bottom, resulting in a gradual and continuous deceleration in both the axial and whirl components of velocity. In a high-head unit such as the one under consideration, a low specific-speed runner is used, so that the velocity to be regained in the draft tube is relatively small and, therefore, the draft-tube design might be considered of little importance. For the load at maximum efficiency this may be more or less true, but when operating such a unit at part loads, there is a considerable whirl component of velocity which is lost in draft tubes of the curved type, but efficiently regained in the spreading draft tube, resulting in a considerable gain in efficiency.

A much more important consideration, however, is that of vibration. With common designs of draft tube, the whirl components at part gates produce eddies and vortices which cause changes of flow in the draft tube and similar changes through the runner and in the penstock. These velocity changes occur with great frequency and result, in the case of long penstocks, in considerable instantaneous pressure variations and a resulting shock in the unit. This is not an unusual condition in existing plants in this country. It is frequently found that in high-head plants where the turbines are provided with curved draft tubes, vibration in the unit is noticed at certain gate openings when there is an absolutely steady load on the turbine, and under these conditions a pressure gage attached to the casing will show rapid variations in pressure of 15 to 25 lb. per sq.in. The use of the spreading draft tube recovers the whirl components of velocity and therefore prevents the formation of the eddies and vortices, resulting in steady flow and consequently an entire lack of vibration or pressure change in the unit.

Casing Design

It was necessary, for shipping reasons, to split the casing of this unit into three sections, these sections being connected by heavy flanges and bolts in order to prevent leakage. In usual designs a separate speed-ring casting is bolted to the upper and lower flanges of the casing, this speed ring consisting of an upper and lower crown connected by heavy speed-ring vanes cast integrally with the crowns. The purpose of this speed ring is to stay the throat of the casing. When such a speed ring is used and the casing is made in two or more sections, it is only possible to carry the bolts on the radial joints into the speed-ring flanges. Hence, there is a considerable length on the radial plane near the throat of the casing which is unsupported, with the result that very heavy loads are concentrated on the bolts. For high-head units such as the one under consideration, this heavy load often results in stretching the bolts near the speed-ring flanges, causing a leakage. This difficulty was obviated in the Southern California unit by casting the speed ring or stay vanes integrally with the casing. By using special care in the steel foundry and by annealing the casing sections, the initial strains in these vanes are removed, and by this means it is possible to carry the bolts in the radial flanges almost all the way to the throat of the casing. Fig. 5 shows a view of this casing in the shop and clearly indicates the position of the speed-ring vanes and the bolting on the radial flanges. Fig. 6 is a general view in the shop, of the casing erected with the Y-pipe connection for the relief valves, the head cover, operating ring, guide vanes

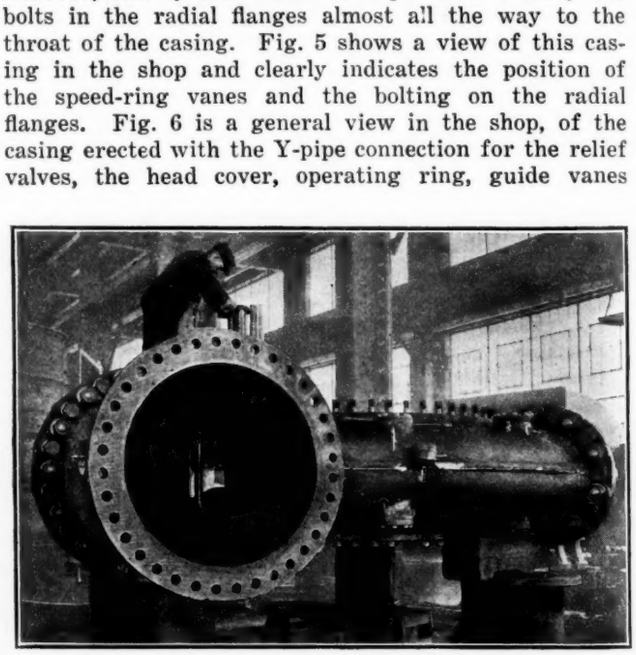


FIG. 5. VIEW OF CASING, SHOWING POSITION OF SPEED-RING VANES

and levers. This casing was tested in the shops to a hydrostatic pressure of 600 lb. per sq.in. and under this pressure showed no leakage whatsoever.

Relief-Valve Design

For high-head units having long penstocks very excessive pressure rises will occur when loads are thrown off the unit unless the relief valve functions properly. For the unit under discussion the penstock is 2,800 ft. long, having a diameter in the upper sections of 7 ft. and in the lower sections of 6 ft. If full load

is thrown off the unit and the governor is adjusted to close the gates in three seconds it is calculated that a pressure rise of 838 ft. will occur at the turbine casing. Fig. 7 shows the changes in pressure at the lower end of the penstock resulting from opening or closing of the turbine gates. These curves were calculated by formulas originated by N. R. Gibson[1] and R. D. Johnson,[2] which formulas the writer believes give the only accurate method for calculating pressure changes.

As already stated, it will be noted from this curve that for full load off the turbine and with the relief valve not in operation, the maximum pressure rise will amount to about 838 ft., which is more than 100 per cent increase over the normal head. The curves are based on a governor time of three seconds, and if the governor were adjusted for a quicker time of operation, the pressure rise would be even more severe. The importance of a reliable relief valve for such units is therefore self-evident. Fig. 8 shows the relief valve furnished with this unit, and in Fig. 1 is indicated the manner in which this valve is connected to the unit. The relief-valve disk is mechanically connected through a piston rod, heavy links, lever and reach rod to the operating ring on the turbine-head cover. When the turbine gates are at rest, the valve disk is held in the closed position by means of the pressure on the bottom of the large balancing piston contained in the cylinder immediately above the relief-valve elbow. This balancing piston is larger in diameter than the valve disk and hence keeps the valve tightly closed with practically no leakage. When the turbine gates close, the relief valve is forced open by means of the turbine operating cylinders through the direct mechanical connections described in the foregoing. In order that the relief valve may act as a water-saving device, an oil dashpot is interposed between the balancing piston and the links. This dashpot has a small bypass connection around the piston and

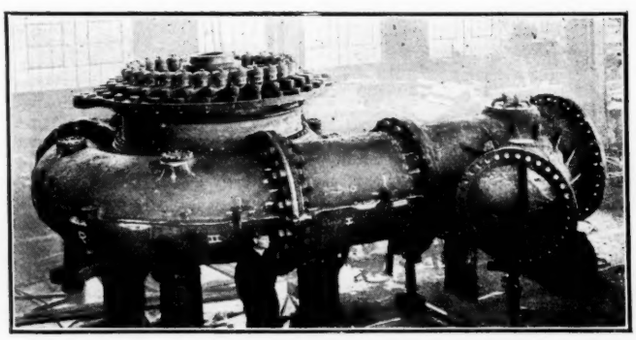


FIG. 6. VIEW OF CASING, SHOWING "Y" PIPE CONNECTION FOR RELIEF VALVE

is provided with a needle valve for adjustment. For quick-closing movements of the turbine gates the dashpot therefore does not come into action and the relief valve opens in exact synchronism with the gates. After the gates have closed, however, the small bypass on the dashpot permits the relief valve to close at a sufficiently slow rate to prevent excessive pressure rise. It will be noted by referring to Fig. 7 that if the needle valve on the dashpot is adjusted for full stroke of the relief valve in 30 sec., the pressure rise due to the relief valve closing amounts to only 70 feet.

In some former designs of relief valves this direct mechanical connection is not used and the relief valve is operated by means of a pilot valve connected to the turbine operating ring, which relieves the pressure under the balancing piston and permits the valve to open. With such designs there is considerable danger of a

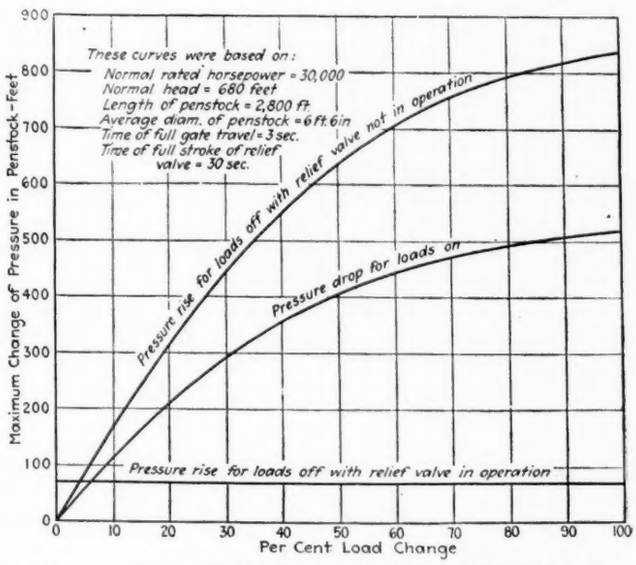


FIG. 7. CURVES SHOWING CHANGE OF PRESSURE AT LOWER END OF PENSTOCK

failure of the relief valve to open in case the small pilot valve becomes jammed or the water passages leading through the pilot valve become clogged with dirt. There is also the danger in such designs of an obstruction becoming lodged below the relief-valve disk, preventing it from opening. Under any one of these conditions, if the load is dropped on the unit, the turbine gates are closed by the governor without a corresponding opening of the relief valve, resulting in excessive pressure rise and danger of a failure of the turbine casing or the penstock.

With the mechanically connected valve provided for this unit, these dangers are entirely eliminated, for there are no small valves to become jammed or clogged and in case an obstruction in the relief valve itself prevents the valve from opening, the turbine gates are at the same time prevented from closing excepting at the very slow rate at which the oil in the dashpot can be bypassed through the small needle valve. All parts of the mechanical connection between the relief valve and the turbine operating ring are designed of sufficient strength to take care of the full load from the turbine operating cylinders, and it is therefore believed that this design of valve is as nearly foolproof or accident proof as can be designed.

Lubrication of Internal Parts

The only internal moving parts in a turbine of this type which require lubrication are the guide-vane shanks. It will be noted from Fig. 1 that for each guide vane three bearings are required, one immediately below the vane, the second immediately above the vane and the third near the top of the guide-vane shank below the lever connection. The two lower bearings operate under a high hydraulic pressure in the

[1] "Pressure in Penstocks Caused by the Gradual Closing of the Turbine Gates," Proceedings of the American Society of Civil Engineers, April, 1919.

[2] Discussion of N. R. Gibson's paper by R. D. Johnson, Proceedings of the American Society of Civil Engineers, August, 1919.

particular unit under consideration, this pressure amounting to 150 lb. per sq.in. The usual method of lubricating these bearings has been by means of small independent grease cups. Under high pressures it is very difficult to force grease into such a bearing by means of a grease cup, and even though a wrench be used on the cap of the grease cup, it is uncertain as to whether any grease is forced into the proper bearing. On a turbine containing 30 guide vanes there would be required 90 grease cups, and from the viewpoint of the operator there is considerable labor involved in the constant filling of these many small grease cups. There is no way of determining whether these bearings are obtaining any lubrication without dismantling the entire turbine, and the unit may be operated for a period of one or two years or more before there is an opportunity to dismantle the unit and make an inspection of these bearings. The consequence is that considerable wear occurs at the guide-valve bearings, which results not only in a leakage of water at these points, but also in lost play in the operating mechanism, causing hunting of the governor.

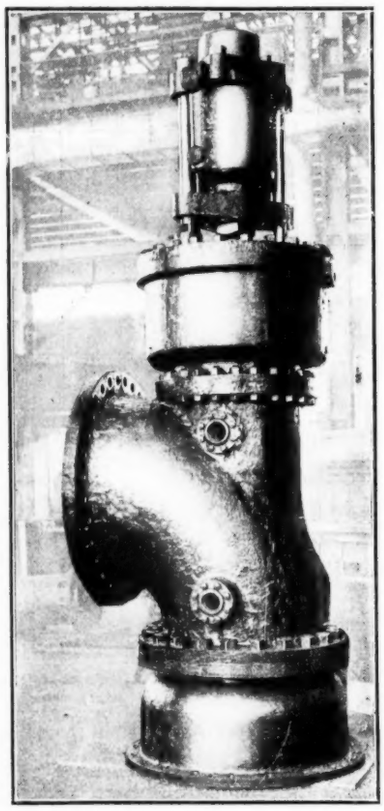


FIG. 8. RELIEF VALVE

To overcome these difficulties in the case of the Southern California unit, the Taylor pneumatic lubricating system is installed for the guide-vane bearings. This consists of a large air-operated grease gun connected by piping to each guide-vane bearing, a separate valve being located in this piping at each bearing. In lubricating these bearings, therefore, it is only necessary for the operator to admit the air pressure to the grease gun and then to open and close, one at a time, each of the 90 valves. With a pressure of 200 lb. per sq.in. on the grease gun this method gives a positive assurance of a proper amount of grease being fed into each bearing. The lubrication of all the bearings is thus easily accomplished without physical effort by the operator, and it is only necessary to refill the large grease gun at infrequent intervals. By reason of this definite and positive means of supplying grease to the bearings, wear at these points is reduced to a minimum.

Governor Control

The governor furnished with this unit is of the I. P. Morris double-floating-lever type No. 2, incorporating a number of new features developed by the author. This governor is supplied with oil under pressure from the motor-driven rotary pumps and is connected by suitable piping to the accumulator tank and the two operating cylinders of the turbine. Fig. 9 shows a view of the governor assembled in the shops. It is of heavy and powerful construction in order to harmonize with the unit it controls. The hand-control mechanism is mounted on the governor base and is shown in Fig. 9 as a small pedestal on the left-hand side. No matter how carefully the governor may be designed and built, there is always danger of trouble occurring due to lost motion, pilot valve sticking, governor belt breaking or other causes, and under these conditions it is always desirable for the operator to be able to take the unit off governor and place it on hand control as quickly as possible.

With former designs this change over from governor to hand control usually entails the closing of three or four large governor valves and the opening of three or four smaller hand-control valves under high pressures. This takes considerable time, and in case of an accident to the governor where the operator becomes excited, there is always a chance of manipulating these valves in wrong sequence and thus entirely losing control of the unit which is something to be avoided if possible.

The governor in Fig. 9 is furnished with an automatic device known as the Taylor control system, consisting of plunger valves in the governor base stand operated by oil pressure from the governor system. A single movement of the lever shown near the top of the hand control pedestal automatically operates all the valves at the same time, closing the governor valves and opening the hand-control valves. This permits a change over from governor to hand control to be made in the fraction of a second and also avoids all danger of runaway or difficulties due to loss of control of the unit during the shift over.

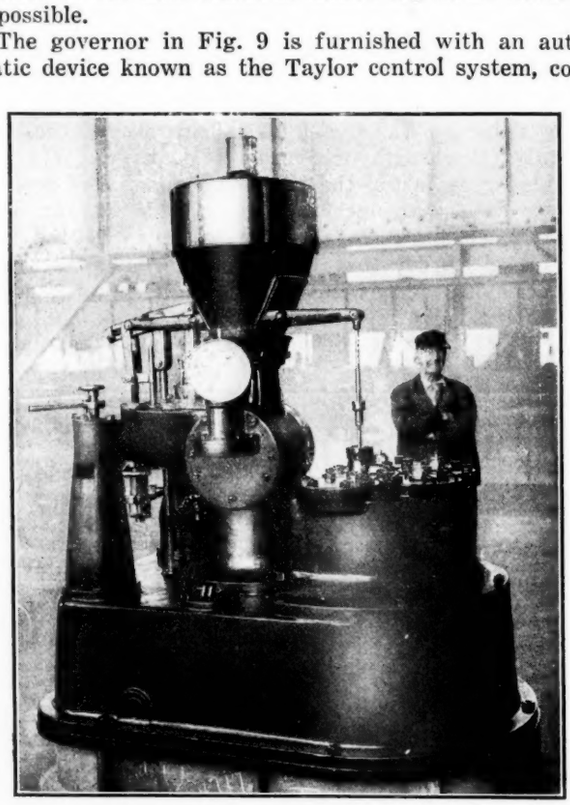


FIG. 9. VIEW OF GOVERNOR ASSEMBLED IN SHOP

A Dream Engine

By Charles Hurst

In odd moments I have planned how I would design a reciprocating steam engine if I had a perfectly free hand, a nice little drawing office at my command and a well-equipped factory capable of turning out work of the highest class. I am not now very much concerned with reciprocating steam engines, as my professional work runs in other channels, but I was brought up on the stationary steam engine, and I retain an old affection for this fascinating and, in some respects, mysterious prime mover. Here, then, is my "dream engine," and if some of the ideas are extravagant, bear in mind it is but a dream, that if the suggestions are not commercially practicable they may lead to healthy discussion and possibly may set others dreaming in a similar fashion. Sometimes the dreams of today become the realities of tomorrow.

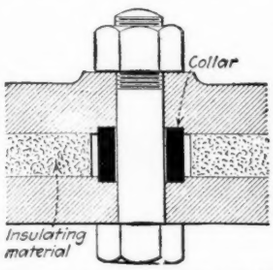


FIG. 1. HEAT INSULATION OF CYLINDER BEARINGS

In making my ideal engine, I shall design for a stop-valve pressure of 250 lb. and a superheat of about 120 deg. F., and whether it is a unaflow or the ordinary four-valve engine does not matter very much, as the ideas are applicable to each type. In the case of the four-valve engine I would make a horizontal four-cylinder triple, with cranks at right angles; that is, there would be a high-pressure cylinder and one low-pressure cylinder tandem on one crank, and the inter

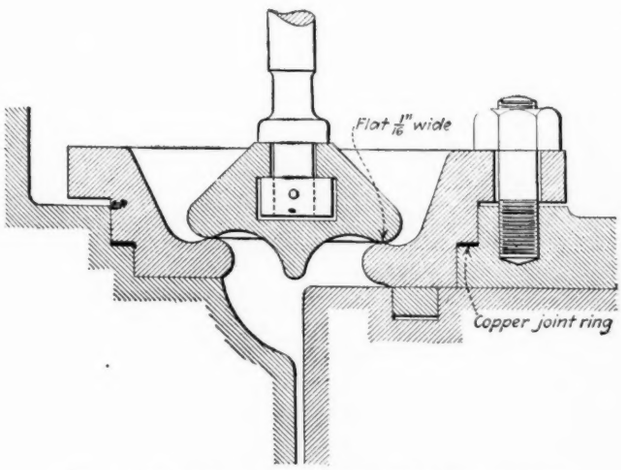


FIG. 2. POPPET VALVE FOR "DREAM" ENGINE

mediate and other low-pressure tandem on the other crank, speed about 200 r.p.m. If I built a unaflow engine, it would have two cylinders with cranks at right angles. Then, when the engine was built, I would erect it in a lofty power house, free from dust, with a tiled floor, a beautiful switchboard, glass office for the chief, baths for the operating staff, and I would appoint a very kind, experienced old chief, gray with years and knowledge, and a stout band of loyal enthusiastic boys to help him.

I should make the heat insulation as perfect as possible. Where the cylinders were bolted to the bed or foundation, I would employ collars so as to allow a thick sheet of non-conducting compound between them, as shown by Fig. 1. The cylinder laggings would be carefully provided throughout and would not be allowed to touch the hot metal at any point. But the big features would be in the valve gear and cylinders. In my opinion there is only one really tight form of valve

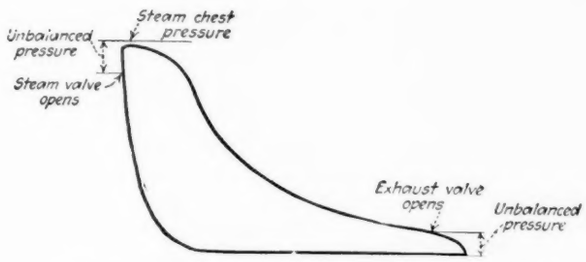


FIG. 3 SMALL UNBALANCED PRESSURE ON POPPET VALVES

in existence. That is the single-faced poppet valve, such as employed on internal-combustion engines, and this type, with certain modifications, I should adopt. Fig. 2 shows my ideas. It will be noticed that both valve and seat are rounded to give a smooth steam flow, the top of the rounding on both valve and seat being slightly flat for a width of about $\frac{1}{16}$ in. With this form of valve and seat I should not hesitate to calculate on a steam velocity of 500 ft. per sec. through the seats, and by this means have small valves and small clearance. All surfaces exposed to the expanding steam in the cylinder would be smooth machined, and in the case of exhaust valves there would be very little condensing surface exposed. In the exhaust valves I should adhere to present practice for area of valve opening. By reason of the small clearance and the high compression employed, the steam valves would open with very little effort, as will be understood by a reference to the diagram, Fig. 3.

The valve gear would be controlled by a powerful inertia governor mounted on the side shaft, varying the throw and position of the steam eccentrics in accordance with the load. Rolling levers would be used for both steam and exhaust valves, so that at the instant of opening the valve gear would have a favorable mechanical leverage, and there would be no reaction of the governor.

The engine would be fitted with ball bearings throughout. In the crankshaft this would effect an improvement in reducing the distance from center of crank in to center of bearing in a direction parallel with the axis of crankshaft. Ball bearings on each side of the cranks would be provided, and the cranks would be forged solid with the shafts. The connecting rods would be made of high-tension steel, of I-section to reduce weight, and by careful design they could be cut down to at least half the weight of an ordinary rod and the running would be improved. Balance weights would be attached to the crank webs. The piston rods would be made of special nickel steel to reduce the diameter to a minimum, thus cutting down heat losses and leakage.

Finally, the whole engine would be of the highest finish throughout. There would be no soft packings whatever. The lubricators and fittings would be nickel plated, and the bed painted with bronze-green enamel. A steam-flow meter, numerous thermometer pockets,

speed recorder, water-measuring tanks and every appliance for exact testing would be included.

Steam consumption? Well, I should hope to reach about 11½ lb. per kw.-hr. and maintain it for many years.

Are there any other engineers who have had their dream engines?

Reports, Records and Meters

By W. W. Edson

There is a large field for improvement and growth in the use of meters and reports around the smaller power plants. These do not actually save money or coal any more than druggist's scales write prescriptions, but they are of value in indicating or measuring losses and gains; in measuring what is produced and what is sold, thus enabling the cost and selling price to be determined; in showing what, when and why something took place, and in showing that the men made the proper inspections and repairs. A few of the purposes of the more common meters and their recording charts are as follows:

A graphic kilowatt meter shows the time and extent of peak and valley loads, time and duration of interruptions of service, serves as a check on watt-hour meters and is used for the purpose of showing when to run different equipment.

A graphic steam-pressure meter is a check on the firemen, especially at night. It shows the time and duration of interruption to the steam supply and registers the steam pressure. The efficiency of the entire plant from the operating standpoint depends somewhat on an even steam pressure.

Graphic feed-water thermometers show under what condition the best temperature was obtained and serve as a check on the performance of the heater and the feed pump. The boiler-plant efficiency depends on the feed-water temperature. The temperature chart is used in getting the average of the day.

Graphic feed-water flow meters are used in making efficiency calculations and to check the performance of the heater and the feed pumps.

A graphic draft gage is used to check the condition of baffles, soot on the tubes, proper draft, frequency and manner of firing, etc.

Grahphic flue-gas thermometers are used in calculating combustion data and to check the proper amount of air, condition of the baffles, soot on the tubes, scale in the boiler, etc.

Flue-gas analyzers are used in calculating combustion data, to check the presence of air leaks and are especially useful in deciding upon the proper method of firing, the thickness of the fire to carry, the amount of draft, etc.

The plant-capacity factor equals the average power for 24 hours divided by the rated capacity of the plant. The higher this factor becomes, the less effect the fixed charges have on the cost of generating electricity per kilowatt-hour.

The maximum-demand factor equals the maximum 15-minute load divided by the rated capacity. This shows if the proper size units are available.

Pounds of steam per kilowatt-hour indicates the efficiency of the turbo-generator. Pounds of water per pound of coal indicates the efficiency of the boiler and furnace. Pounds of coal per kilowatt-hour indicates the combined efficiency of the plant.

A list of reports and records to be kept in a 1,000-kw. plant or smaller is suggested as follows as a guide to those interested:

Man Responsible for Report	Name of Report	Items Included on Report
Fireman....	Coal.....	Pounds, bushels, or loads of coal burned per shift
Engineer...	Blue.....	Condition and cleanliness of plant, machinery, and fires at beginning and end of watch. Meter readings Supplies and oil used or needed Repairs made or suggested Interruptions of service and causes Hours run for each unit
Chief.....	White.....	Time and overtime for each man Total daily meter readings Amount of coal on hand Report on service interruptions, repairs made or needed, and supplies used or needed
.....	Station.....	If required by State Public Service Commission Similar to white report
Chief.....	Supplies....	Date box received, shipper, order number, condition and contents
Chief.....	Charts	
Chief.....	Permanent..	A record for each machine, showing name and address of maker and shipper, date received, cost, cost of installing, size and rating, dimensions, weight, location in plant, plant number, and data relative to pipe or wire sizes, connection diagrams, material in various parts, sizes of packing, tubes, etc., and any other information needed
Chief.....	Operating...	One sheet for each unit. Shows date and output. Repairs. Changes. Cleaned. Inspected. Oil changed.
Lineman...	Work cards.	Time and supplies used on each job
	Meter cards.	Information relative to meter tested
Bookkeeper	.....	Necessary reports and records for State Public Service Commission. Correspondence file catalogs Timebook Bill file File of information on all contracts, etc., in the safe File of permanent plant data File of complaints made by customers File of accidents File of data and curves calculated, as required (see list above) Monthly reports showing kilowatt-hours generated Kilowatt-hour commercial lighting Kilowatt-hour industrial power Kilowatt-hour city lighting Kilowatt-hour city power Kilowatt-hour plant power Kilowatt-hour plant lighting Kilowatt-hour lost Kilowatt-hour total sold

REVENUES

Item.....	Amount.....	Per Kw. Sold
Power.....	.....	.....
Lighting.....	.....	.....
City.....	.....	.....
Waterworks.....	.....	.....
Miscellaneous.....	.....	.....
Total.....	.....	.....

EXPENSES

Item	Amount	Per Kw.-Hr. Gen.	Per Kw.-Hr. Sold
Coal.....	.....	.....	.....
Oil.....	.....	.....	.....
Supplies.....	.....	.....	.....
Maintenance and repairs.....	.....	.....	.....
Water.....	.....	.....	.....
Labor.....	.....	.....	.....
Total operation...	.....	.....	.....
Insurance.....	.....	.....	.....
Interest.....	.....	.....	.....
Depreciation.....	.....	.....	.....
Total expenses....	.....	.....	.....
Net income.....	.....	.....	.....
Extension.....	.....	.....	.....
Outstanding debts.	.....	.....	.....
Surplus or deficit..	.....	.....	.....

Obviously, the expenses should be charged on the books to such items as power, transmission, distribution, consumption, commercial, general, undistributed, depreciation, interest, etc.

Curves should be plotted from day to day showing the daily total kilowatt-hours generated and the load factor which equals the average power for 24 hours divided by the maximum load for 15 minutes. The larger this item the more efficient the plant becomes. A shift of a load from one part of the day to another may raise this factor. It indicates the percentage of reserve power of the plant which must be kept constantly available.

The Plant That Died of Improvements

By John S. Carpenter

"Say Pop!" cried Jimmy, shaking the old Chief's hand with great vigor, "tell us where you've been and what you've been doing! You've been away so long that I'll bet the boys who read *Power* think you an' me is dead."

The grizzled old warrior of the hydro plant peered over his glasses with eyes full of gay malice at the boy. He handed Jimmy a duplicate unit to the one he had already lighted and seated himself in his old cushioned chair, grunting with satisfaction to be back on the old job.

"Well sir, I've been up to Squantumville on that new Tukkyhoe plant. It sure is a daisy, but I guess they're goin' to close 'er down. She's died of improvements."

"Of what?" Jimmy regarded him with mirthful doubt. It was some seconds before he could bring his

"AN' EVERYTHING GOES ALONG SMILIN' UNTIL THE BIRD SUDDEN-LIKE CASHES IN"

wits to focus on the old man's thought. "Well, Pop, that sure is a new one on me. You'll have to explain it a whole lot better than that."

"Huh!" Pop grunted. "Ain't you heard tell of patients that has been operated on fer this and fer that, an' ain't the doctor's bulletins showed 'great improvements' after the thing was did? An' everything goes along smilin' until the bird sudden-like cashes in!"

Jimmy nodded, seeing a great light. He had just noticed that Pop's mustache was gone.

"Now, that there plant ain't such a big one, an' between you an' me it could have been put in fer about half the price it really cost them. But they got a new hydraulic engineer down the main office, an' him bein' one of them up-to-the-minute on everything new in this business, he sets to work an' puts up a model plant. He advertises fer bids on the machinery equipment an' everything. He gets all them cute salesmen in his parlor an' they talk him deef, dumb an' blind. I seen him after one of them birds just let him loose an' you'd a thinked that he had just took dope. 'It's gonna be great, Pop,' he tells me, '2 per cent here, 5 per cent there, a 90 per cent turbine an' a peach of a new-fangled draft tube.' An' old timer like me has to give in when they springs that kinda news. I'm too far behind the times."

"Is it all true, Pop?" asked Jimmy, a bit dazed by the big talk.

"Well son, when I sees how the engineer succumbed to that kinda mental chloroform, I guesses as how I better take, turn to, an' learn somethin' of the new improvements. I asks him if he'd mind coachin' me up a little, an' he bein' one prince of a chap, he says, 'Sure thing!—and what's more I want you in on the next conference—I want you to know what's going on—you're the boy that's going to run the plant.' That was more than I really wanted. After that I felt as if I'd owed him somethin'; he sure bought me out that time!"

"What are the hydraulic conditions there, Pop?" Jimmy was now so interested that he puffed absent-mindedly at a cigar that was cold and stale.

"Why it's 2,000 hp. in two units under 25-ft. head, the generators runnin' at 157 r.p.m."

"Oh, I see! I thought it was something pretty big." Jimmy woke up and lighted the cigar.

"Well, I got in the next conference with the bird who I was tipped off on the quiet was to get the contract. He could get a talkin' machine rattled. When they started the victrola goin', I did nothin' but listen. I asked myself if all them luxuries was goin' to cost them much, or was it one of those bids that this here turbine factory was goin' to give us the benefit of their brains on to get the job. Pretty soon I felt myself slippin'. That salesman pulls out a lotta tests made on an experimental plant, a much lower head than ours. He says, 'If this thing works as good as this on a model-sized plant, how much better it'll work on your full-sized plant!' I asked him, 'We like this dope all very fine, but what will this cost us compared with the ordinary standard type of equipment?'

"The look he gave me was the kind you hand anybody that's just sprung somethin' terrible. He says: 'My dear sir, you want to get the very most out of your water rights—you want every drop that goes through the turbine to sweat blood giving up all the power there's in it—and you want the very last word in equipment in order to accomplish this. You want every kilowatt there's to be gotten to send out over your lines and save your company all the coal you can.' He subsides after that an' sees it sink in on me.

"'Now,' I says to our engineer, 'how do the other bids stack up with this man's? Can't the other people give us the same stuff?'

"There was the salesman's cue. 'Why no!' he snaps, 'not at all! This apparatus is all covered by letters patent—we've got about twenty on this thing alone—got it so sewed up that our competitors couldn't get in with a shoehorn!' After that I kept still, but I was aboilin' over inside o' me. That bird had his say and he went out.

"I sat with our man fer the rest of the afternoon. We went over all the bids—had to forget two of 'em because they were offerin' the ordinary type of unit like we got in this plant. They just proposed the square-box open-flume setting with the old-style steel draft tube, which would have satisfied me under those low-head small-power conditions, but the engineer says 'No!' with a big N. It all lay between two bids. Both had all the latest trimmin's, but our engineer's choice was the one offered by the human victrola; because he told us confidentially that his firm was gonna bring suit on the other man for patent infringement. He scared our man stiff when he asks him if we want to get in on the suit an' get an injunction out on us."

"I suppose, Pop, that spilled the beans," declared Jimmy, seriously.

"Yep," snapped Pop, "that cooked the pie. That was almost a year and a half ago; everything is runnin' fine. They's no question at all but what they fulfills all their guarantees."

"But—but where's the hitch, Pop?" asked Jimmy, waiting for the climax.

"Huh!" the old Chief grunted, "first place the 90 per cent turbine cost us 35 per cent more than the standard 87 per cent machine; the turbine builder was sure goin' to play safe so he'd meet his guarantees, so he saddles our old power company with the cost of cutting down the velocity in the penstock; makin' fancy curved concrete work on the racks so the water wouldn't lose much head in goin' through them; buildin' a pretty spiral case in concrete an' makin' us excavate good an' deep in the tailrace, besides a expensive-formed concrete draft tube on which we pays a gold-tasseled royalty per horsepower; all of which costin' us more than we can sell the juice for an' meet competition."

"Ha, Ha!" Jimmy laughed, comprehending.

"Then, you sees, it never rains but it pours! That new power company three miles downstream puts in a new plant with standard equipment, designs their work conservative an' has us beat sellin' juice. We figured out all our costs an' are findin' out that they underbid us 15 per cent for power an' ain't half sold out yet. Their units don't give no 90 per cent efficiency, but they sure do give 'way up in the eighties, an' I betcha in less than two years from now ours won't be doin' any better. Squeezin' that last 2 per cent outa the water is tall pushin'."

"What does our engineer say?" asked Jimmy.

"He ain't said much a-tall, but I've gotta hunch that when he gets any ideas like that again on a small plant, he's agonna whisper it to the transmission towers an' listen to what they sings in the wind.

"Also, son, mind yer don't think I'm one of them kind which don't believe in the best of generatin' equipment, but here's when it don't pay to go into all them there refinements."

Jimmy studied the thought for a few moments. "And they're going to shut the plant down?"

"Not quite, son, but the company's pretty sure they can sell the plant to that new automobile factory who'll be glad to get it."

A Quick Shaft-Repair Job in a Textile Printing Works

BY Z. BRADSHAW

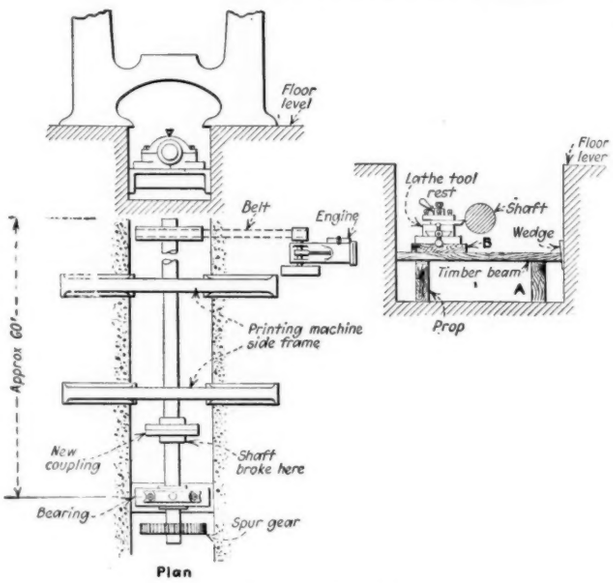
At a large textile printing works a serious breakdown occurred some months ago. A fracture was discovered in the cast-iron main driving shaft, rotating below the floor level and centrally underneath the whole battery of printing machines. This break, fortunately, took place in close proximity to a bearing, otherwise the job would have presented greater difficulties.

The plan view indicates the position of the shaft at the broken section. Owing to the congested space in which the accident happened, the works engineer was confronted with a difficult problem, because two important factors were to be considered. One was how to withdraw the longest portion of shaft without dismantling a large quantity of complicated and nicely adjusted machinery; the other was how speedily to effect the repair so that none of the factory plant would be forced to shut down.

An outside engineering firm quoting for the repair job—that is, the operation of taking out and fitting a new 14-ft. length of shafting—estimated that it would take at least seven days. This long period of waiting was unthinkable. The works engineer solved the problem in the following manner:

The small length of broken shaft with the spur wheel keyed on was removed to the repair shop and necked down in the lathe to a smaller diameter to receive half of a new flanged coupling. The second operation was to rotate the long length of shaft without removing it from its bearing and to receive the companion half of flanged coupling. First of all two timber beams were fitted with props and secured by means of wedges into the wheel race as shown at *A*.

A cross girder *B* was placed on these two beams and on the girder was fitted a lathe compound tool rest. All



LAYOUT OF SHAFTING AND WAY OF TURNING END OF SHAFT

that was now necessary was to find some method of rotating the long part of the broken shaft. A 4-hp. steam engine was procured and the belt connected to the end of the lineshaft as shown. The engine was held to the floor with convenient weights and connected to the mill steam supply. The shaft was then faced on the end and turned down. As the machining was in progress, the engineer in charge was busy chasing about for a new coupling. Fortunately, a local company had one in stock. Cutting the required keyway in the shaft and in the coupling and fitting the latter in position finished the repair, which was completed 34 hours from the time of the breakdown, which luckily occurred at the end of the week. By putting in night work on the job everything was ready for Monday morning, much to the surprise and satisfaction of the management.

An economical use for exhaust steam is found in Berlin, where all the hot water for certain public baths is obtained from an electric-power plant three-fourths of a mile away.

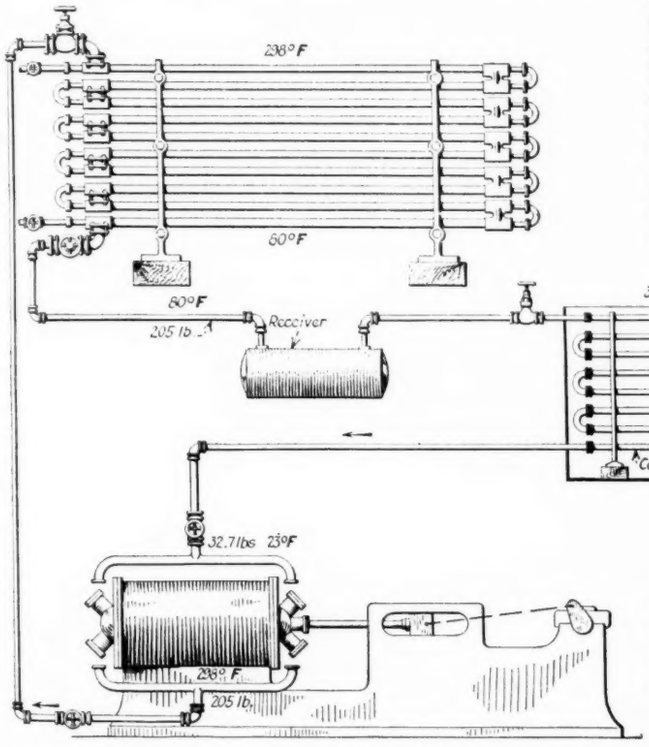
Some Practical Refrigeration Problems

By C. H. HERTER

IN DISCUSSING various refrigerating problems with a new assistant engineer in a large New York ice plant, the question of the proper rate of water circulation over the cooling tower and atmospheric ammonia condensers, also the work required from the compressor, came up. The problems were worked out in detail and may possibly be instructive to others who desire to know the "reasons why" in mechanical refrigeration. As it would take considerable space to analyze the design and performance of condensers, only the item of the quantity of heat to be abstracted per ton of work done is gone into at this time.

The Engineer's Problem

The engineer propounded this question: "How many heat units must be removed from one pound of ammonia gas at atmospheric presure and 80 deg. F. to liquefy it?"



TEMPERATURE AND PRESSURE CONDITIONS IN A REFRIGERATING SYSTEM

His own answer was 628.12 B.t.u., while by the writer's calculations the lowest figure was 643.3 B.t.u.

In ordinary refrigerating plants the question as put is not likely to come up because the liquefying within the condenser is being done by the combined action of pressure much above atmospheric and the removal of heat by cooling water. No attempt is made in practice to liquefy ammonia under a compression pressure of 14.7 lb. absolute (0 gage). Under this low liquefaction pressure the boiling temperature (using Keyes & Brownlee's ammonia tables, 1916, which include the properties of superheated ammonia, De La Vergne Machine Co. catlog) is —28 deg. F.

Using first cold water and then other ammonia at some temperature colder than —28 deg. F., it would be possible to remove the 80 — (—28) or 108 deg. of superheat and the latent heat at —28 deg. from the pound of ammonia. Referring to the tables, it will be noted that Q, the heat content of ammonia vapor at 14.7 lb. absolute with 100 deg. superheat, is 588 B.t.u. When the ammonia is dry saturated—that is, with no superheat—the total heat content is 537.5 B.t.u. Thus the heat of superheat is 588 — 537.5, or 50.5 B.t.u. per pound for 100 deg. of superheat, which means that the specific heat of superheated ammonia vapor in this range is 0.505 B.t.u. per degree F. For the total 108 deg. of this problem it would be 54.5 B.t.u. To this must be added the latent heat of evaporation L at 14.7 lb. pressure and —28 deg. F., equal to 607.3 B.t.u. Consequently, liquefying a pound of ammonia at 14.7 lb. absolute pressure and superheated 108 deg. will require a removal of 607.3 + 54.5, or 661.8, B.t.u. If one makes use of Bureau of Standards ammonia tables, the latent heat of evaporation at 14.7 lb. is 588.8 B.t.u. The heat to be removed is then 588.8 + 54.5, or 643.3 B.t.u. Thus the difference in results of the calculations is only 18.5 B.t.u., or 2.4 per cent. Until the Bureau of Standards tables include the properties of superheated ammonia, one may work with two sets of tables to receive greater accuracy, using the Bureau of Standards tables for saturated ammonia and Keyes & Brownlee's or Goodenough's superheat tables. Turning from the foregoing hypothetical case to a more practical example as met with in the ordinary plant, we determined the amount of heat to be removed by cooling water per ton of refrigeration done at 20 lb. gage evaporator pressure and 190 lb. gage liquefaction, or condenser pressure.

At 34.3 lb. absolute (19.6 lb. gage pressure) within the ammonia coils in the freezing tank the saturation temperature is, by Bureau of Standards tables, 5 deg. F. Owing to excessive length of pipe coils per feed valve, long or poorly insulated mains or too small a suction pipe, there may easily exist a presure drop between compressor and coils of, say, 1.6 lb., giving a compressor suction pressure of 32.7 lb. absolute and a saturated temperature of 3 deg. F. A further loss caused by superheating of the suction vapor in the suction line by heat absorption from the outside, say 20 deg. above 3 deg., may exist, resulting in a compressor-inlet temperature of 23 deg. F. The various temperatures and pressures are given in the illustration, a diagrammatic compression system.

Within the cylinder the absolute pressure at start of compression curve may be 32 lb. and the discharge pressure 205 lb., resulting in a ratio of compression pressure $\frac{205}{32}$, or 6.4 : 1.

With the usual exponent of the compression curve for ammonia vapor, 1.3, and assuming that the temperature of the suction vapor is at least 23 deg. F., the theoretical discharge gas temperature with adiabatic compression is

obtained by means of the following well-known equation:

$$\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \frac{T_2}{T_1}$$

Substituting the values of P_1 and P_2 and remembering that T_1 and T_2 are absolute temperatures—that is, degrees Fahrenheit plus 460—we obtain as the following value of T_2 if n is taken as 1.3:

$$(23 + 460) \times \left(\frac{205}{32}\right)^{0.23} = 741 \textit{ deg. absolute} = 368 \textit{ deg. F.}$$

Depending upon design and speed of compressor and on the absorption of heat from the cylinder and valves the actual discharge temperature observed may be as much as 19 deg. above, or 298 deg. F., equivalent to 200 deg. of superheat above the saturation temperature of 98 deg. at 205 lb. absolute pressure. The temperature rise of 19 deg. is chosen for convenience to allow of using directly the values appearing in the tables of superheat. The rise may be much more, depending on the cylinder design.

The heat content Q of ammonia vapor at 200 deg. superheat and 205 lb. absolute pressure is (K. & B. tables) 669.35 B.t.u. per lb. at 0 deg. superheat, or at the saturation state, it is 556.55 B.t.u. Hence, the superheat to be removed in the first part of the condenser 669.35 — 556.55 = 112.8 B.t.u. per pound of ammonia vapor, the mean specific heat being 0.5625 for this range of temperatures.

At 98 deg. F. the latent heat of evaporation (Bureau of Standards tables) is 480.2 B.t.u. per lb., this being the work of the second, or liquefaction, part of the condenser. If the condenser used is of the counter-current type and has ample cooling surface, it is capable of cooling the condensed liquid to, say, 80 deg. F., thus removing (see heat content of liquid) 152.8 — 132, or 20.8 per pound.

The total work of water on the condenser is, therefore, as follows:

Extracting superheat	112.8 B.t.u. per lb. =	18.4 per cent
Liquefying	480.2 B.t.u. per lb. =	78.3 per cent
Liquid cooling	20.8 B.t.u. per lb. =	3.3 per cent
	613.8	100.0

Useful Refrigerating Effect

The useful refrigeration obtained in this plant per pound of liquid ammonia equals the total heat of saturated ammonia vapor at the coil pressure, 34.3 lb. absolute and temperature 5 deg. F., which is 613.8 B.t.u., minus the heat already in the liquid as it passes the expansion valve; that is, heat of the liquid at 80 deg. F., or 132 B.t.u.

Thus, the heat available for useful work in the freezing tank is 613.8 — 132 = 481.8 B.t.u. The loss due to 20 deg. superheating in the sucton main represents 10.3 B.t.u. per lb and has nothing to do with the refrigerating effect in the coils. It is, however, included in the 18.4 per cent of work done in the condenser because it has added 20 deg. to the initial ammonia temperature.

Cooling at the rate of 2,000 lb. ice melting per 24 hours is equivalent to removing heat at the rate of 200 B.t.u. per minute, being one ton of refrigeration. In the plant under discussion it is, therefore, necessary to circulate ammonia at the rate of $\frac{200}{481.8} = 0.416$ lb. per minute per ton of refrigeration, and since in most large plants it requires only 1.5 tons refrigerating effect per ton of ice harvested, $0.416 \times 1.5 = 0.624$ lb. liquid must be circulated per minute per ton of ice. Since the volume of vapor at 32.7 lb. absolute and 23 deg. temperature is 9.184 cu.ft. per lb., 0.624×9.184 or 5.73 cu.ft. (9,900 cu.in.) of suction vapor will have to be pumped per minute per ton of ice sent to ice storage. If the true volumetric efficiency of the compressor happens to be 75 per cent, then the net piston displacement required is 9,900 ÷ 0.75 = 13,200 cu.in. per ton of ice, or 8,800 cu.in. per ton of refrigeration.

The work of the condenser water per ton of refrigeration is the heat units removed per pound times the pounds of ammonia circulated per ton, $613.8 \times 0.416 =$ 255 B.t.u. per minute; and if, as usual with recooled condenser water, the difference between outlet- and inlet-water temperature is 10 deg., and since one pound of water will absorb 1 B.t.u. per one degree temperature rise, 25.5 lb., or 3.06 gal. of water must be circulated per minute per ton of refrigeration, equivalent to 4.6 gal. per ton of ice. To allow for water loss and radiation to the atmosphere, these amounts may be increased 5 per cent. A greater rate of water circulation than 6 gal. per min. per ton of ice is not advised unless the saving in compressor horsepower is greater than the additional pump work involved.

Heat Equivalent of Power

The heat removed by the condenser water is seen to be 255 B.t.u. per minute per ton of refrigeration under the temperatures assumed, while the useful cooling effect is only 200 B.t.u. The difference, 55 B.t.u. represents the heat equivalent of the power expended within the compressor cylinder. The heat equivalent of one horsepower per minute is $\frac{33{,}000}{778}$, or 42.4 B.t.u. per minute; and $\frac{55}{42.4} = 1.3$ indicated compressor horsepower per ton of refrigeration. This figure checks fairly well with 1.336 hp., the value given on York Manufacturing Co.'s data table C48 for their single-acting compressors, published in the *Journal* of the American Society of Refrigerating Engineers, May, 1915.

Many otherwise well-informed engineers are under the impression that the condenser water has to absorb only the 200 B.t.u. per minute per ton of refrigeration abstracted by the heat-absorbing pipes. They forget that the work done in the compression of the vapor appears as heat in the discharge vapor, being a loss incidental to the refrigeration process, which must also be taken up by the cooling water. Since the water jacket of the compressor cylinder has inadequate surface, its capacity for absorbing heat from the gas in the short space of time available is very small.

To obtain maximum efficiency, a plant should be so constructed that the coldest water is utilized first for subcooling the condensed liquid as low as is compatible with the cost of the additional cooling surface required. In this way the heat of the liquid ammonia as it leaves the condenser already mentioned may be reduced. So long as manufacturers strive to cut down the condensing surface to a minimum and pay no attention whatever to the important factor of liquid cooling or precooling, so long will the economy of refrigerating machines remain below the maximum. In his article on "Standard Unit of Refrigeration," in *Power*, April 5, 1921, page 559, the writer showed that for every 4 degrees subcooling of condensed liquid the output of a machine is increased about 1 per cent, and the horsepower per ton is reduced.

Effects of Voltage Fluctuations on Direct-Current Motors

Limitations in Voltage Variation—Comparison of Motor Characteristics When Lowering Voltage on Both Field and Armature—What Happens on a Motor When the Voltage Fluctuates Ten per Cent from Normal—Both Interpole and Non-Interpole Machines Are Considered

By H. M. PHILLIPS
Electrical Aid, Public Works Department, Navy Yard, Washington, D. C.

INASMUCH as practically all motors are subject to voltage fluctuations of more or less severity, their behavior under these conditions is a matter of considerable importance, although rarely referred to in either the manufacturer's or the purchaser's specifications. In the handbook of the Electric Power Club it is stated that all motors except fractional-horsepower machines "shall operate successfully at normal rated load at any voltage not more than 10 per cent above or below the name-plate rating, but not necessarily in accordance with the standards of performance established for operation at normal rating." The reader may define "successful" operation to suit himself, doubtless the manufacturer will do likewise. It is doubtful if the writers of the specification intended it to apply to a sudden variation from 10 per cent below to 10 per cent above normal, or vice versa. In fact, it would be hardly fair to attempt to apply it in this manner, as the action of the motor will depend largely upon the nature of the load, which is not usually a factor that is under the control of the motor manufacturer.

Data on Motor Operation

Some motors, however, are much better suited than others for operation under fluctuating voltages, the difference depending upon their ability to carry very heavy overloads for short periods and upon the variation of motor speed with load. Unfortunately, it is seldom that exact information on either of the foregoing points can be obtained prior to the purchase of the motor and actual trial by the purchaser. The full-load speed is given with a fair degree of accuracy by the manufacturer and in some cases the no-load speed may also be obtained (both at rated voltage). However, intermediate speeds and more especially overload speeds do not vary uniformly with load, and speed at voltages other than normal is not usually obtainable.

Even though all the desirable data are obtained on a particular motor, its usefulness is decidedly limited, for another of the same type and manufacture but of different horsepower or speed may show quite different characteristics, while even a motor that is supposed to be an exact duplicate may have sufficient variation to affect materially its performance with a fluctuating voltage. The reason for this difference may be ascribed to slight variations in the magnetic saturation of various parts of the armature and field, which are likely to be due to the quality of the iron, also to slight variations in the air gap (which is frequently adjusted by the manufacturer in order to obtain the rated speed) or to variations in brush setting, either accidental or intentional. After the motor has had some service, the condition of the commutator is also likely to have a considerable influence upon the speed.

Curve *A*, Fig. 1, may be taken as a rather common type of motors of from 5 to 15 hp.; larger motors are likely to show less speed variation, especially on overloads. It is to be noted that there is a considerable drop in speed between no load and 25 per cent load, after which the curve is comparatively flat up to full load, and that there is an increasingly rapid fall of speed as the overload is increased. The load in all cases, for the purposes of this article, is measured as a percentage of full-load amperes. This general type of curve is found for the majority of shunt motors, with or without interpoles. The actual differences in speed, however, are subject to very considerable variations as is also the amount of overload that can be carried before the overload drop begins. Interpole motors are sometimes found in which the speed rises with an overload (Fig. 3) although if sufficient load can be carried the overload droop must finally assert itself. A motor which on any part of its speed curve shows an increase of speed with load is to be avoided for nearly all kinds of service.

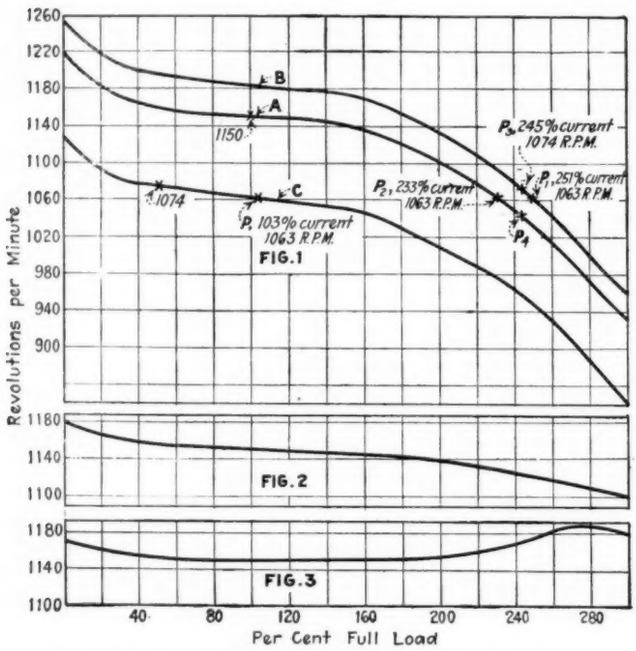
Effects of Reducing Field Voltage

Curve *B* shows the change in curve *A* that may be expected to result from a 10 per cent reduction in field voltage, the armature voltage remaining constant. The distance that *B* will lie above *A* depends on the saturation of the magnetic circuit which varies very materially with different motors; a 3 per cent speed increase for the larger part of the curve may be taken as fairly common, with slightly lower values on overloads.

Lowering the voltage of both field and armature by 10 per cent should give curve *C*, which lies approximately 10 per cent below curve *B*. The effect of armature and interpole-field resistance is to lower it still farther, but is so small that it may safely be disregarded except on heavy overloads. In all cases the speed of the motor will fall off with a decrease in voltage, but never in direct proportion to the voltage.

As an illustration of the effect of sudden changes in voltage it is convenient to assume that the motor is running at full load and normal voltage with a very heavy flywheel for the load. A centrifugal drying machine, which has the effect of a flywheel, may serve as a concrete example. If the voltage has a sudden drop of 10 per cent, there can be no appreciable change of speed during the decrease. However, the full-load speed of curve *A* is higher than the no-load of curve *C*, therefore the motor will act as a generator and supply current back into the line. In extreme cases, where there is a larger drop in voltage or where the speed curve is unusually flat, this reverse current may exceed the full-load current and trip circuit breakers or blow fuses. In this and the following cases, however, it will

be assumed that circuit breakers are provided with a "time element" which will prevent their being thrown except by prolonged overloads. The reverse current will act as a brake, reducing the speed of the load until the no-load speed of the curve *C* is obtained, when the motor will begin to take current in increasing quantity as the speed is reduced until a point is reached where the current is sufficient to prevent further loss of speed.



FIGS. 1 TO 3. LOAD-SPEED CURVES FOR DIRECT-CURRENT MOTORS

If it is assumed that the same torque (pull exerted by the armature) is required at the lower speed, the current will exceed its original value, on account of reduced field strength, by about the same percentage as the increase in speed between curves *A* and *B* and the speed will become constant at the point *P*.

If the voltage now returns very suddenly to its original value, the armature current will respond much more rapidly than the field, on account of the induction of the latter and the motor will be momentarily under the conditions of curve *B*. The speed of the load, not having time to change, the current will rise momentarily to the value shown at P_1. But as the field is rapidly strengthening, probably in a fraction of a second, the motor will change to the operating condition of the point P_2 on curve *A*, where the heavy overloaded current taken by the motor will gradually increase the speed until the original conditions are re-established. In this case the maximum load is shown to be 251 per cent of the full-load current, which is a rather severe strain on the commutating ability of any motor.

The Standardization Rules of the American Institute of Electrical Engineers require a motor to carry a momentary overload of 100 per cent, and it is probable that the additional load referred to in the foregoing will be successfully carried if not of too long duration. In the case under consideration the load will drop very rapidly to 233 per cent, where it reaches curve *A*, after which the rate of decrease will depend principally upon the inertia of the moving parts of the driven system. Increasing the horsepower of the motor is by no means a sure method of overcoming the troubles arising from voltage fluctuations. It is not unlikely that the larger motor will have a flatter speed curve, which will more than counterbalance the greater power available for acceleration. In actual practice the extreme overloads obtained in the foregoing would be considerably modified; the rise in voltage, though frequently rapid, is not instantaneous. The heavy overload will in itself tend to hold the voltage down, and any induction in the circuit, including that of the armature and interpole field, will tend to prevent the armature current from rising quite as rapidly as the voltage.

On the other hand, many motors have speed curves less favorable for service on fluctuating voltage than those used in the illustration. Motors with commutating poles are frequently found to have very flat speed curves, as shown in Fig. 2, while occasional cases show a curve similar to Fig. 3. It is evident that if commutation is maintained and the load continues to rise, the speed curve must finally fall, but the motor is pretty certain to flash over before that point is reached. Even the shunt motor without interpoles will sometimes show a remarkably flat speed curve. The writer recalls one instance in which repeated measurements showed the full-load speed to be somewhat higher than the no-load, with the speed at half load slightly below either. Overload speeds were not obtained, but this motor would suffer severely on a fluctuating voltage.

Work Required for Acceleration

The work required for acceleration varies to a startling degree with the diameter and revolutions per minute of the body being accelerated; as illustrated by Fig. 4, and the following table:

FOOT-POUNDS OF WORK REQUIRED TO INCREASE SPEED OF ROTATING BODIES WEIGHING IN EACH CASE 710 LB.

	Bodies Shown in Fig. 4					
	A	B	C	D	E	F
R.p.m. 1,063 to 1,150.....	49,000	20,000	18,200	10,100	6,600	3,000
798 to 862.....	27,500	11,250	10,230	5,680	3,700	1,680

The rotating element of a centrifugal fan as ordinarily constructed has a rather large diameter and high speed of rotation as compared to the majority of other machines requiring the same power; the inertia is therefore rather high. The centrifugal pump with approximately the same power and speed will generally have a rotor of smaller diameter and lighter weight than the fan and, having less inertia, is less susceptible than the fan to voltage fluctuations. Both the diameter and the speed of either fan or pump may vary considerably with the pressure for which they are designed and the individual preference of the designer, which may reverse the foregoing statements in some instances, although they will be found to hold true for the majority of cases.

Although the inertia of the moving system is the most important factor governing the performance of a motor under fluctuating voltage, the trouble may frequently be aggravated by a fluctuating load. The most conspicuous example of inertia is probably the centrifugal drier. However, it gives a load that is practically constant. Punch presses and similar machines are equipped with flywheels that supply the greater part of the force required to work the metal. It can exert this force only through a reduction of speed at the moment the work is being performed. This reduction in speed increases the load on the motor until normal speed is restored, but the momentary force required is so great that it may be assumed as derived entirely from the wheel and is independent of the actual force exerted by

the motor. As an illustration assume that the motor and press are so proportioned that when the former is operating on curve *A*, Fig. 1, the maximum load on the motor is 160 per cent and falls, as the flywheel accelerates, to a minimum of 50 per cent. This gives maximum and minimum motor speeds of 1,160 and 1,136 r.p.m. respectively, or a change of 24 revolutions, which is sufficient to allow the flywheel to do the work. On curve *C* a maximum speed of about 1,074 r.p.m. would occur and if the sudden rise to normal voltage takes

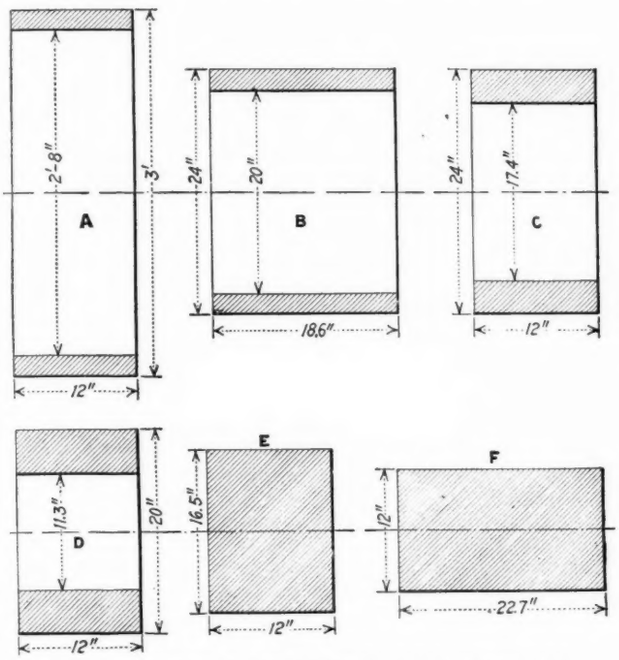


FIG. 4. SECTION THROUGH BODIES REFERRED TO IN THE TABLE

place at the same instant that the flywheel is called upon for its energy, it must first jump to the point P_3 on curve *B* and then drop 26 revolutions to the point P_4 on curve *A* before acceleration can begin, which materially increases the duration of the excessive overload. If the motor had a flatter speed curve, the additional overload, small in the present case, might be a very serious factor, with a characteristic such as Fig. 2. Even a slight fluctuation in voltage might be sufficient to cause the motor to flash over, while with the curve, Fig. 3, the motor would undoubtedly behave badly even at constant voltage. Many machines that are not ordinarily equipped with a flywheel are dependent on the inertia of the armature and other rotating parts of the system to give the necessary energy to prevent momentary overloads of considerable magnitude and are troublesome with fluctuating voltages. Among these may be mentioned reciprocating pumps, planers, shapers and other machines having reciprocating parts. In some cases a pair of gears that do not mesh perfectly at all points, but which run fairly well with a motor operating at constant voltage and having a falling speed curve, may produce trouble with a motor having a flatter curve and operating on a fluctuating voltage.

An adjustable-speed motor running on weak field at high speed will show a smaller percentage of change in speed for a given variation in voltage than is obtained by a constant-speed motor, or with the same motor while running on full field. Adjustable-speed motors are, as a rule, able to carry a heavier overload on full field than on weak field, the net result being that the adjustable-speed motor is generally most likely to give trouble on a fluctuating voltage when running at its highest speed. Constant-speed motors that are wound for high speed are also rather susceptible to changes in voltages both on account of the inertia of their armatures and the fact that they usually have a flatter speed curve than the medium- or slow-speed motor of the same capacity.

Commutation will also be slightly more difficult with a voltage increase than decrease, on account of the increased voltage between commutator bars. Fortunately, voltages above normal are of less frequent occurrence than below.

Variations in Motor Efficiency

The motor efficiency (output ÷ input) does not vary greatly with a variation in voltage within 10 per cent of normal and may be either raised or lowered. A properly designed motor should have its maximum efficiency at approximately its rated voltage, but various practical considerations may cause it to lie considerably above or below this. The variation in efficiency, however, is so small as to be negligible unless the cost of power is unusually high. The change in speed produced by a change in voltage has in nearly all cases a decidedly important bearing on the efficiency of the plant as a whole. If the speed obtained at reduced voltage is all that is needed, the increased speed at normal voltage means a very considerable waste of power. In the case of a centrifugal fan or pump the power varies approximately as the cube of the speed; if 10 hp. is required to drive at 1,070 r.p.m., it will take approximately 12.4 hp. when driven at 1,150 r.p.m., a direct loss of 2.4 hp. For nearly all classes of work the minimum increase of power will be in direct proportion to the speed which would give, for a 10-hp. machine, a loss of at least ¾ hp. while running at the higher speed; in most cases the loss would be considerably greater. On the other hand, a reduction in the speed of machinery used for manufacturing purposes generally means a corresponding reduction in the output of the plant, which is likely to be a very serious matter, although it often escapes immediate detection.

Voltage fluctuations of frequent occurrence are in all cases a serious handicap on a plant, in the way of decreased efficiency and general wear and tear on motors and machinery, even though acute motor trouble is not experienced. In cases where it is impracticable to remedy the trouble, the evil effects may sometimes be minimized by resorting to one or more of the following expedients:

1. The use of slow-speed motors.
2. The use of motors with a rapidly falling speed curve. Improvement in this respect may sometimes be obtained by a slight change in brush setting or by the use of a shunt on the interpole winding.
3. Decreasing the inertia of the load on troublesome motors. This, unfortunately, is seldom practicable.
4. The use of compound-wound motors in preference to shunt-wound. This is merely another way of producing the falling speed curve and is of great advantage in a large number of drives where a shunt motor is commonly employed, either on account of a slight difference in first cost or through ignorance of the superiority of the compound winding for the service in question.

Reduction-Gear Applications

By IRA SHORT
Engineering Department, Westinghouse Electric and Manufacturing Company

APPLICATIONS of the double helical gears are now used to a great extent in connection with high-speed turbines. Actual tests have proved that the efficiency of a single-reduction gear, as shown in Fig. 1, is as high as 98.5 per cent. The double-reduction gear shown in Fig. 3 has an efficiency of 97.5 when transmitting 1,500 shaft horsepower and reducing from 3,360 to 90 r.p.m. This efficiency was obtained with oil having a viscosity of 280 sec. Saybolt at 100 deg. F.; with an oil having a viscosity of 560 sec. the efficiency was 94.2.

The life of a correctly cut gear is practically indefinite if the teeth are maintained in alignment and properly lubricated, as the teeth never come in contact, being separated at all times by an oil film. Accurate cutting of teeth and the maintaining of alignment between the teeth is of the greatest importance in gears of high tooth pressure and high pitch-line speed.

Most Efficient Speed

Reduction-gear development has greatly increased the useful field of the steam turbine. The most efficient speed of the turbine is usually much higher than the most efficient speed of the apparatus that it is required to drive. By interposing a reduction gear, both the driving and driven apparatus are allowed to operate at their most economical speed and a much higher over-all efficiency results. Also, the higher-speed turbine for the same power is smaller and a more sturdy and reliable machine.

Turbines are designed for the highest speed that they can be and direct-connected to standard apparatus. For instance, 3,600 r.p.m. corresponds to a 60-cycle two-pole generator. The same turbine without any change can be used to drive a 25-cycle generator at 750 r.p.m. or a 40-cycle generator at 600 r.p.m. by driving through a gear. A 750-kva. geared alternator is shown in Fig. 2.

Direct-current generators necessarily operate at slow speed to insure good commutation, the speed varying from 514 r.p.m. for 1,500-kw. to 1,200 r.p.m. for 500-kw. units. In Fig. 1 is shown a 1,500-kw. unit, the turbine speed being 3,600 r.p.m. and the generator speed 514 r.p.m.

Steam Turbine's Starting Torque

A steam turbine's starting torque is about twice the full-speed torque. This is approximately the same as an electric motor. It is not uncommon to use an electric motor for driving lineshafting. Turbines are also operated for similar purposes. In one installation a turbine operates at 3,600 r.p.m. and transmits through a double-reduction gear 600 hp. at 103 r.p.m. to the lineshafting of a paper mill. In another installation a geared turbine drives a lineshaft through double-reduction gears and transmits 500 hp. from 5,744 to 150 r.p.m. At another plant the reduction from 3,000 to 140 r.p.m. is made in one step and the power transmitted is 1,700 horsepower.

Gears are not always utilized to reduce the speed; often they are used to increase speed. At the Redlands Irrigation Co., Grand Junction, Col., a hydraulic turbine running at 360 r.p.m. drives a centrifugal pump at 700 r.p.m. and transmits 600 hp. The Anaconda Copper Co., Anaconda, Mont., has a 3,500-r.p.m. blower driven through a gear by a 2,250-hp. motor running at 1,759 r.p.m. A gear is often used for connecting a direct-current motor to a high-frequency generator. In one unit recently built, the motor speed was 1,700 r.p.m. and the generator speed was 6,300 r.p.m.

Low-head centrifugal pumps must operate at low speed to be efficient. The pump is generally designed to be driven direct by an electric motor or by a steam turbine through a gear. In order to make the unit compact, the gear and turbine are sometimes combined as shown in Fig. 7, or they may be separate. These drives are also much used for driving fans or blowers, and quite recently a number of gears like that shown in Fig. 4 were installed with electric motors for driving the rolls of a paper machine. For small power the drive shown in Fig. 5 is compact and therefore economical of space.

A rather unique gear installation is that of a 3,375-hp. turbine which is connected through a gear to two gas-compressor cylinders. The gear shaft is extended on either end to which the compressor cranks are fitted. The speed of the turbine is 1,500 r.p.m. and the gas compressor makes 120 double strokes per minute, or 60 revolutions per minute.

Reduction Gears in Marine Work

The largest application of the reduction gear is no doubt for marine propulsion on account of the saving in weight of a geared turbine over a direct-connected turbine. The gear can be designed to meet any space or power requirement. Where floor space is limited, the pinion can be placed on top, as shown in Fig. 3. This gear is 1,500 shaft horsepower, and reduces from a turbine speed of 3,360 r.p.m. to 90 r.p.m. of the shaft. On the larger sizes it is usual to divide the turbine, making a high-pressure turbine and a low-pressure turbine. The gear is then made with two pinions, as shown in Fig. 6. The gear transmits 3,000 shaft horsepower, reducing from 3,360 r.p.m. to 90 r.p.m. The pinions may be placed at any angle to the horizontal that will best fit in the given space. In Fig. 3 the pinion is on top, while in Fig. 6 they are in a horizontal plane.

Four gears are now being built in which four pinions mesh with each gear wheel. Two of the pinions are about 45 deg. above the horizontal center line and two are about 45 deg. below the horizontal center line. Two high-pressure turbines are connected aft of the gear to the pinions below the center line and two low-pressure turbines are connected forward of the gear to the pinion above the center line.

The applications briefly described have actually been installed or are being built. New gear applications are continually developing. The Diesel engine is now receiving much attention for ship propulsion. The maximum power for which a Diesel engine can be built to run satisfactorily is limited. The field of the Diesel will no doubt be broadened by gearing two or more engines to a single propeller.

FIG. 1. SINGLE-REDUCTION GEAR BETWEEN TURBINE AND DIRECT-CURRENT GENERATOR

Generator 1,500-kw. capacity operating at 514 r.p.m.; turbine runs 3,600 r.p.m.

FIG. 2. SINGLE-REDUCTION GEAR APPLIED TO TURBO-ALTERNATOR

Turbine operates at 3,600 r.p.m., generator runs at 750 to 600 r.p.m., depending on the frequency.

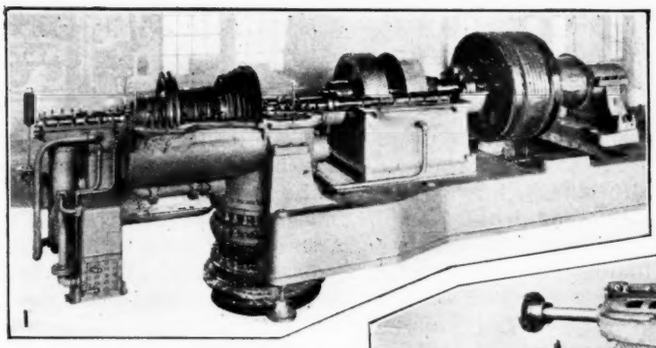
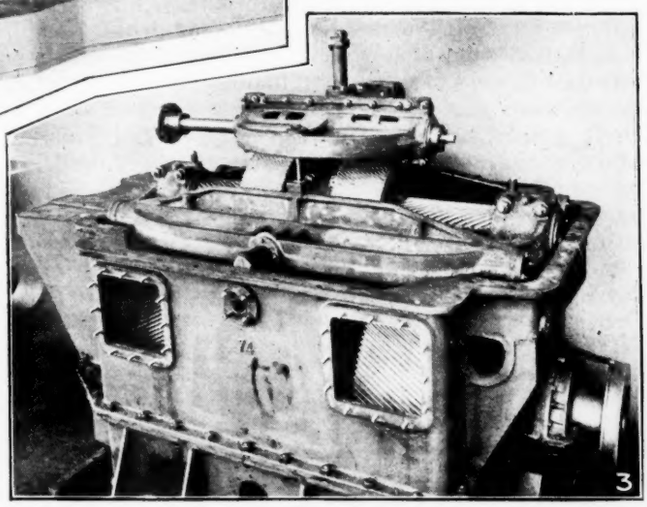
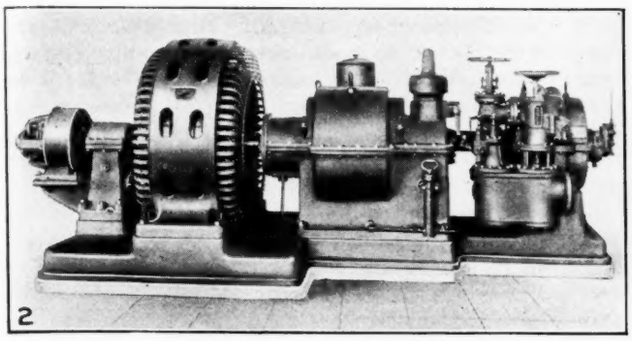
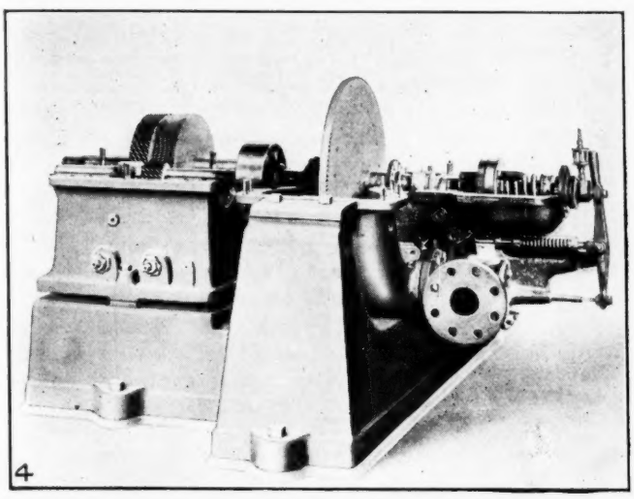


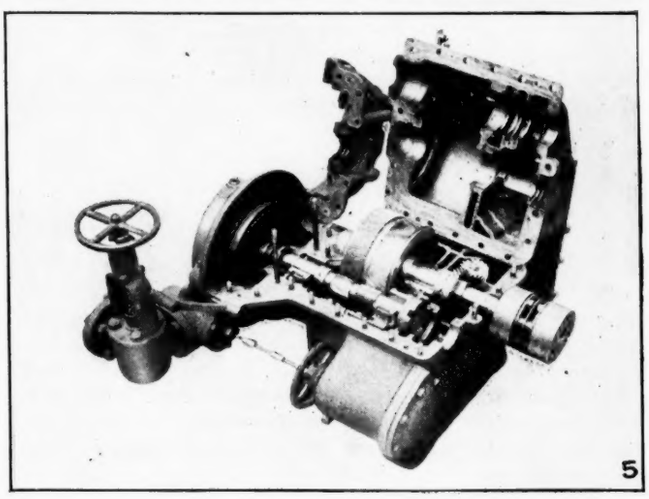
FIG. 3. DOUBLE-REDUCTION GEAR WHICH TRANSMITS 1,500 SHAFT HORSEPOWER AT 3,360 R.P.M. TO 90 R.P.M.

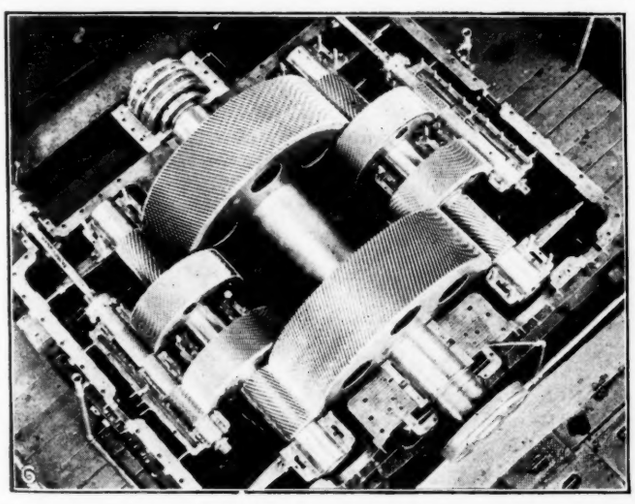
The pinions in this application are placed above the gears.

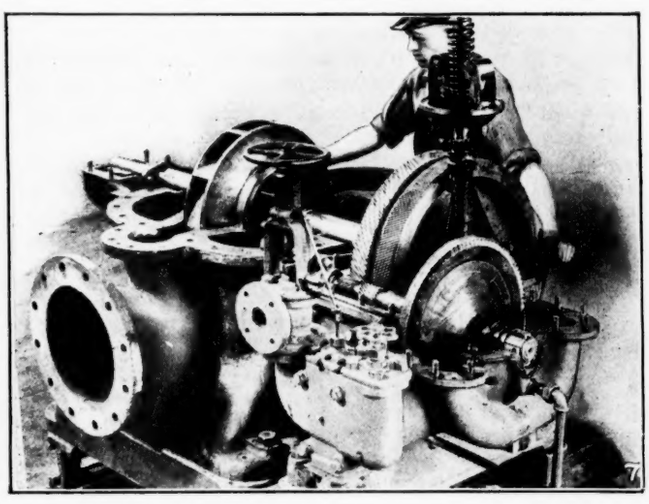












FIGS. 4 TO 7. SINGLE- AND DOUBLE-REDUCTION GEARS

Figs. 4 and 5—Turbine and reduction gears for small power. Gears of this class have been used with electric motors for driving to rolls of paper machines. Fig. 6—Double-reduction gear with pinions in horizontal plane. Gear transmits 3,000 shaft horsepower, reducing from 3,360 to 90 r.p.m. Fig. 7—Turbine, reduction gear and slow-speed centrifugal pump combined into one unit.

Cutouts for Motor Protection

By EDGAR P. SLACK
Assistant Electrical Engineer, Underwriters Laboratories

THE Code of rules on the overload protection of motors cover both the location of the cutouts in the motor circuits[1] and the sizes of cutouts to be used; the latter will be discussed briefly in this article.

When an electric motor is subjected to a heavy load, it doesn't "lie down on the job," but gets down to work and does its best to carry it. In doing so, it necessarily draws more current from the supply line, and this heavy current heats the motor windings. It is, therefore, necessary to provide the motor with a cutout of sufficiently low rating to prevent overheating, which might burn out the motor or shorten the life of the insulation.

Motor manufacturers, in assigning current ratings, are concerned chiefly with the heat that will be produced in operation. Motors designed for continuous service are marked with a "continuous" rating, showing the current they can carry indefinitely without getting too hot. Many machines are intended for short-time service, consisting of runs alternating with stops of sufficient length to allow considerable cooling. Motors for elevators, cranes, steel mills, ore and coal-handling machinery, and the like may be mentioned as examples of short-time machinery. These machines are given "short-time" ratings of five minutes to two hours, representing the length of time that the machine can be operated at full load without exceeding its temperature rating.

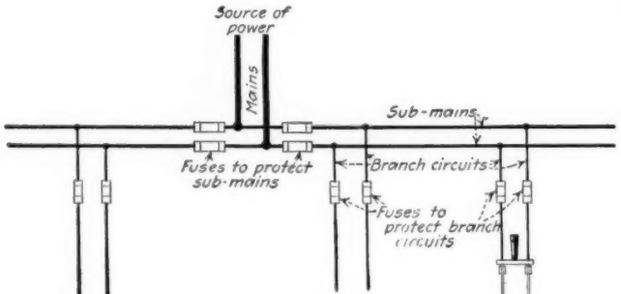


FIG. 1. ILLUSTRATING DIFFERENCE BETWEEN FUSES FOR LINE AND MOTOR PROTECTION

Many of the Code rules on fuse sizes for motor installations deal with the protection of the line supplying the motor rather than the protection of the motor itself. The fuses installed for protection of the line should by no means be confused with those installed for the protection of the motor. The capacity of the line fuses is determined by the size of the wires composing the line, whereas that of the motor fuses is determined by the capacity of the motor. The protection of the line will be discussed fully later, but a typical example is shown in Fig. 1. This diagram represents an ungrounded circuit with mains from the source of power feeding sub-mains, which in turn supply the branch circuits to the motors. The sub-mains, which are smaller than the mains, are fused where they join the mains; and the branch circuits, which are smaller than the sub-mains, are fused where they join the sub-mains, the size of the fuses in each case being only slightly larger than the carrying capacity of the line to be protected. The object of these fuses is to blow in case of a short-circuit or overload on the line, and thus prevent the flow of a heavy current, which might burn out the line or shorten the life of the insulation.

In some cases the cutouts required for the line are considered sufficient for the motor as well. For example, all motors rated two horsepower or less may be installed without motor fuses or cutouts, for the line cutouts are considered sufficient protection for the motor itself. Motors of other than "continuous" rating, or used on other than constant-load duty, such as elevator or crane motors, and others mentioned in the foregoing discussion of "short time" ratings, may also be installed without motor cutouts. Squirrel-cage and similar motors, whose running fuses are cut out of circuit during starting, are likewise considered sufficiently protected by the line cutouts during the starting period.

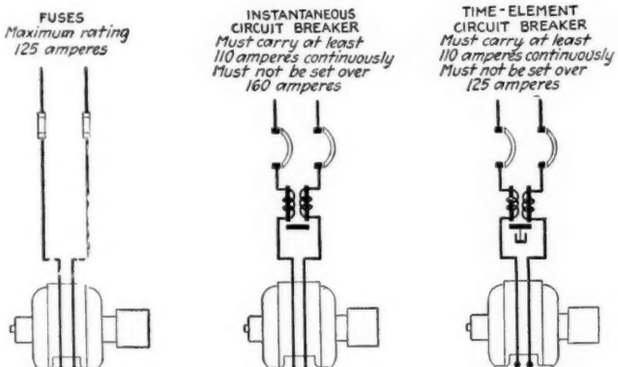


FIG. 2. CORRECT RATING FOR FUSES AND SETTING FOR CIRCUIT BREAKERS FOR MOTOR PROTECTION

The Code requirements for motor protection, represented in Fig. 2, provide that "each continuous-rated motor of over two horsepower used for constant-load duty must be protected by running fuses, thermal cutouts or a circuit breaker in accordance with the following: "If fuses or thermal cutouts are used, their rated capacity must not exceed 125 per cent of the name-plate

What sized fuses should be installed to protect your motors? How should you set your circuit breakers to keep within the Underwriters Rules? This information is given and discussed in this article, the third of the series dealing with the requirements of the National Electrical Code.

[1] See article by the author on Overload Protection of Motors, July 26, 1921, page 132.

current rating of the motor, except that when no fuses or thermal cutouts of the required capacity exist, those of the next higher standard rating may be used." For instance, fuses for a 25-hp. 250-volt direct-current motor having a name-plate rating of 100 amperes should not be over the 125-ampere size. A 10-hp. 110-volt three-phase motor rated 54 amperes would call for 1.25 × 54 or 67.5-ampere fuses, but the 70-ampere size, being the next larger standard size, would be acceptable. This size of fuse permits the motor to be run at 25 per cent overload, it being common practice for continuous-rated motors to have a two-hour guarantee at this overload.

"If a circuit breaker is used, it should have a continuous-current capacity of at least 110 per cent of the name-plate current rating of the motor." This rule provides a factor of safety to prevent overheating of the breaker and is necessary because of the overloads that are likely to occur. For a motor rated 100 amperes, the circuit breaker should be able to carry at least 110 amperes indefinitely.

If the breaker is of the ordinary instantaneous type, which responds immediately to an overload or short-circuit, it may be set up to not over 160 per cent of the name-plate rating of the motor; that is, for a 100-ampere motor the breaker could be set as high as 160 amperes. It is probable that a circuit breaker with this setting would operate sooner than a fuse of the specified rating.

Considerable overloads, if they are only momentary, are not likely to overheat a motor seriously. For this reason time-limit circuit breakers are recognized, but they must be set at a lower value, not over 125 per cent of the motor rating, on account of the reduced degree of protection afforded. This type of breaker, like the instantaneous type, depends for its operation on the magnetic pull exerted by a solenoid on an armature, but has additional parts to retard the operation on moderate overloads, so that if the overloads are of short duration they may pass without the breaker opening. In one well-known make of breaker the pull on the armature is communicated to a smooth-faced disk engaging with a seat covered with oil. As the disk is pulled up, the oil film becomes attenuated and, if the overload is continued long enough, is finally ruptured, allowing the breaker to open in the usual manner.

Preparation Cost of Pulverized Coal*

By P. F. Coffin†

The cost of preparing pulverized coal, naturally, varies with the kind, the moisture content, the size of plant, the load factor of operation, the type of equipment, etc. Muhlfeld[1] states that the smallest size of plant worth considering for pulverizing coal for boilers would have an output of 80 tons a day. Where the competition with other fuels is close, as in burning high-grade bituminous coal on stokers, a 200-ton plant might better be considered as a minimum economical size.

A milling plant having an actual daily output of about 200 tons of coal for use in a metallurgical plant is reported to have an average cost, over a period of six months, of $1.05 per short ton for preparation and conveying. This includes all fixed charges on the investment as well as operation and maintenance. The engineers of another plant with a considerably larger output report that their cost of operation and maintenance ranges from 75 to 80 cents per ton. This includes the cost of handling, preparing, conveying and firing, as well as the cost of tramming the coal to the plant and dumping it, which item is about 10 cents. This leaves a cost of 65 to 70 cents a ton exclusive of the fixed charges on the investment. In both of these plants ring-roll types of mills were used. The figures for the first plant were for 1920, and for the second, for 1919.

These plants were both installed a good many years ago, and it is well to examine an estimate on the cost of pulverizing coal in a modern plant. Two examples are taken, with daily outputs of 500 and 1,000 tons respectively, using 57-in. Fuller mills having a capacity for pulverizing 8 tons of bituminous coal per hour to standard fineness. This is the largest ring-roll mill built for grinding coal. The unit costs for power, labor and drier coal are assumed by the author, the remaining data being furnished by the Fuller Engineering Co.

The following table contains data on the capacity, cost, etc., of milling plants containing four and seven mills respectively.

Number of 57-in. mills installed	4	7
Number of mills in operation (allowing a spare unit)	3	6
Daily output of plant (in tons of 2,000 lb.)	500	1,000
Number of hours required:		
At 7 tons per mill per hour	24	24
At 8 tons per mill per hour	21	21
Size of driers (2 units)	5½ x 42 ft.	6½ x 50 ft.
Size of building	40 x 128 ft.	40 x 160 ft.
Cost of Installation:		
Building	$25,000	$28,000
Machinery	175,000	240,000
Total	200,000	268,000
Annual fixed charges at 15 per cent	$30,000	$40,200
Daily fixed charges for 330 days (assuming half-time Sundays)	$90.90	$121.81

The installation costs are of July, 1921, and include Fuller-Kinyon pumps and 200 ft. of pipe for transporting the coal to the furnace bins.

When operating continuously, the mills will need to grind only about 7 tons an hour for a daily output of 500 tons. An output of 8 tons per mill will reduce the time required to approximately 21 hours daily operation, the remainder of the time being available for repair work by the operating force. With four units in operation, and grinding 7 tons an hour, 500 tons can be ground in 18 hours. With an output of 8 tons an hour only 16 hours will be required. Hence there is a possibility of operating the 500-ton plant on a two-shift basis provided the maintenance work is done on the third shift. It is advantageous to provide a storage bin between the crusher and driers of sufficient capacity to enable the crusher to do its daily work in ten hours.

The operating force is these plants would be made up as follows on each shift:

Output of plant, tons	500	1,000
Number of shifts	2 or 3	3
Miller and helper, each, hours	20	21
Drier fireman, hours	20	21
Crusherman and helper, each, hours	18	16
Additional helper, hours	.....	20
Total labor hours	96	115

When the 500-ton plant is operated on two shifts with four mills running for 16 or 18 hours daily, an allowance of 20 hours is made for contingencies. Hence the millers and fireman would work in two shifts of 10 hours and the crushermen in two shifts of nine hours. By crushing all the coal in 10 hours, as suggested, two crushermen only would be required for a single 10-hour

*From the author's contribution, "The Utilization of Coal on a Multiple Products Basis," to R. F. Bacon's and W. A. Hamor's forthcoming book "American Fuels."

†Research Laboratory, General Electric Company.

[1]"Powdered Coal," by John E. Muhlfeld, Proc. Engineer's Society of Western Penn., May 1920.

shift. The labor required for maintenance is not included in the foregoing, hence it would be possible to employ all the operatives on shifts of equal length by having them do repair work when their machines are not running. This would also apply in the case of the 1,000-ton plant when the miller and fireman and additional helper are employed in three 8-hour shifts.

Labor is figured at 55 cents an hour, including 50 cents for helpers, 65 to 70 cents for mill foreman and a small allowance for superintendence.

The cost of drying will be offset, in part, by the improvement in the net calorific value of the fuel, since the moisture must be evaporated in the furnace when burning raw coal. The usual efficiency of the Fuller drier is about 6 lb. of moisture evaporated per pound of coal fired. The proportion of coal consumed for drying the remainder ranges from 1 to 1.5 per cent when the moisture content is 7 per cent.

OPERATING COST DATA ON COAL MILLING PLANTS

Capacity of plant, tons per day	500	1,000
Equipment, number of 57-inch mills	4	7
Labor hours, daily, in 8-hour shifts	96	115
Drier coal	7	14
Power, 14 kw.-hr. per ton	7,000	14,000
Daily Plant Costs		
Power, at 1.5c. per kw.-hr	$105.00	$210.00
Drier coal, at $7 per ton	49.00	98.00
Labor, at 0.55c per hour	52.80	63.25
Maintenance, at 7c. per ton	35.00	70.00
Fixed charges	90.90	121.81
	$332.70	$563.06
Cost per ton prepared	$0.666	$0.563

If the demand for coal falls off during part of the year, the cost per ton will be somewhat higher, and this will also be true if the plant is operated regularly with a smaller daily output. The following tables indicate the effect of operating the plant at load factors of 70 per cent and of 35 per cent, the outputs in the previous tables being taken as 100 per cent load factor. These factors are based on the operation of the plant with two shifts and with one shift, respectively, of 8 hr. each. Maintenance work that cannot be done during operating hours can be done in the idle periods, and this will allow of more continuous operation during the working shifts. In the case of the 500-ton plant 66.6 per cent labor has been assumed for 70 per cent load factor and 33.3 per cent labor for 35 per cent load factor. In the 1,000-ton plant the percentage of spare equipment is less and a larger percentage of labor hours has been assumed for 70 per cent load factor. In spite of this, 96 labor hours may produce 700 tons in two 8-hr. shifts in the larger plant as compared with 500 tons in two 10-hr. shifts, or in three 8-hour shifts in the smaller plant.

COAL-MILLING PLANTS OPERATED WITH LOAD FACTOR OF 70 PER CENT

Output of plant, tons per day	350	700
Labor hours, assumed (two shifts per day)	64	96
Daily Plant Costs		
Power	$73.50	$147.00
Drier coal	34.30	68.60
Labor	35.20	52.80
Maintenance	24.50	49.00
Fixed charges	90.90	121.81
	$258.40	$439.21
Cost per ton prepared	$0.738	$0.627

MILLING PLANT OPERATING WITH LOAD FACTOR OF 35 PER CENT

Output of plant, tons, per day	175	350
Labor hours, assumed (1 shift per day)	32	48
Daily Plant Costs		
Power	$36.75	$73.50
Drier coal	17.15	34.30
Labor	17.60	26.40
Maintenance	12.25	24.50
Fixed charges	90.90	121.81
	$174.65	$280.51
Cost per ton prepared	$1.00	$0.80

Handy Boiler-Room Formulas

By C. T. Mitchell*

The following formulas may be found useful, although they are not an attempt at anything new, but merely a restatement of fundamental law in what the writer has found to be a convenient form.

The rise in temperature of boiler-feed water due to passing through an economizer may be obtained from the following formula:

$$R = \frac{T_g - T_w}{\frac{W}{AK} + \frac{2W}{CG} + \frac{1}{2}} \qquad (1)$$

R = Rise of temperature of feed water in passing through economizer, deg. F.;
A = Area of heating surface of economizer, sq.ft.;
K = Coefficient of heat transfer, B.t.u. per sq.ft. per deg. F. temperature difference per hour;
G = Pounds of flue gas per lb. of coal;
C = Coal burned per hour, lb.;
W = Water passing through economizer per hour, lb.;
T_g = Temperature of gas entering economizer, deg. F.;
T_w = Temperature of water entering economizer, deg. F.

Formula (1) is derived by assuming that the specific heat of the gases is 0.25 and that the temperature of both water and gas change at a constant rate in passing through the economizer.[1] These assumptions are sufficiently close for most practical purposes. All the values may be known or predicted with reasonable accuracy with the exception of K and G. The heat transfer rate K is the most uncertain quantity and varies from approximately 5 to almost zero, depending on the condition of the tubes, velocity of water and gas, temperature difference, etc. Probably 3 is a conservative average, although higher values are sometimes used. Since K is indeterminate with any high degree of accuracy, the entire formula becomes only an approximation in company with other formulas of the same character. Its chief value is in obtaining rapid comparative values of the temperature rise under varying conditions. Although K has a large influence on the value of R, it will be noted that the temperature rise does not vary directly with the heat-transfer rate.

Starting with an estimated value of either G or T_g, the other can be obtained from one of the formulas:

$$T_g = T_r + 4\frac{HF}{G} \qquad (2)$$

$$G = \frac{4\,HF}{T_g - T_r} \qquad (3)$$

where

T_g = Temperature of gas leaving boiler, deg. F.;
T_r = Room temperature, deg. F.;
H = Heat value of coal, B.t.u. per lb.;
F = Fraction of total heat in coal lost in waste gases from boiler;
G = Pounds of flue gas per lb. of coal.

In applying formula (3) it will be found that G is greater than ordinarily allowed for in designing forced-

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[1] The corresponding drop in temperature of gas $= \frac{WR}{.25\,CG}$

Formula (1) is based on the algebraic mean temperature difference between the water and gas as expressed in the following equation.

$$WR = AK\left[\left(T_g - \tfrac{1}{2}\times\frac{WR}{.25CG}\right) - \left(T_w + \tfrac{1}{2}R\right)\right]$$

draft equipment for underfeed stokers. It must, however, be remembered that the gases issuing from the boiler have been augmented in volume by leaks through the settings. Stack temperatures would ordinarily be higher than they are were it not for this dilution with cold air. Other things being equal, economizer efficiency is seriously affected by dilution, as may be demonstrated by the application of formulas (1) and (2) to specific cases. Of course a slight increase in the value of R may be expected with diluted gas, owing to greater velocity past the tubes. To offset the effect of low gas temperature, however, R would have to increase out of all proportion to its behavior under observation, which indicates an increase in proportion to the square root of the gas velocity.

Problem Showing the Operation of the Formulas

The following problem will serve to show the usefulness of these formulas in checking the performance of an economizer which is already in operation:

A boiler plant with economizers is operating under the following conditions:

Coal fired per hour.....24,000 lb. = C
Water evaporated per hour.....200,000 lb. = W
Heating value of coal.....13,000 B.t.u. per lb. = H
Temperature of gases entering economizer 570 deg. F. = T_g
Temperature of room..... 70 deg. F. = T_r
Temperature of feed water entering economizer .....150 deg. F. = T_w
Assumed fraction of total heat of coal present in flue gases entering economizer = 0.2 (or 20 per cent) = F.
Area of heating surface of economizer = 15,000 sq.ft. = A.
Assumed coefficient of heat transfer for these conditions = 3 = K.

It is desired to compute the rise in temperature of the feed water passing through the economizer and compare it with an actual observed rise of 52 deg. in order to determine whether the performance of the economizer is as good as might reasonably be expected.

To solve this it is first necessary to get G (the pounds of flue gas per pound of coal) from formula (3). Substituting in (3) gives:

$$G = \frac{4\,HF}{T_g - T_r} = \frac{4 \times 13{,}000 \times 0.2}{570 - 70} = 21$$

If the percentage of CO_2 in the flue gas is known, the value of G can be easily figured from it without the use of the foregoing formula. Using G, it is possible to get the theoretical temperature rise (R) of the feed water as follows:

$$R = \frac{T_g - T_w}{\frac{W}{AK} + \frac{2W}{CG} + \frac{1}{2}} = \frac{570 - 150}{\frac{200{,}000}{15{,}000 \times 3} + \frac{2 \times 200{,}000}{24{,}000 \times 21} + \frac{1}{2}}$$

$$= \frac{420}{4.45 + .79 + .50} = 73 \text{ deg. rise}$$

The actual observed rise, however, is only 52 deg. Since the value of K used in the formulas was conservative, it must be assumed that something is wrong with either the economizer or the draft.

To illustrate the effect of air dilution on economizer efficiency, assume that G (the weight of gas per pound of coal) has increased from 21 to 30, owing to excess air, poor settings and leaky flues. The greater velocity of the gases past the economizer tubes produces better heat-transfer conditions and therefore a higher value of K (heat-transfer coefficient). On the assumption that it increases as the square root of the gas velocity, a good estimate for K under this new condition would be 3.6. It will also be assumed that F (fraction of heat in coal lost in flue gases leaving boiler) remains constant at 0.2, although in fact it might be greater in this case. Formula (2) now gives T_g (temperature of gas entering economizer).

$$T_g = T_r + \frac{4HF}{G} = 70 + \frac{4 \times 13{,}000 \times .2}{30} = 416 \text{ deg.}$$

Substituting the new values of G, K and T_g in Formula (1) gives

$$R = \frac{T_g - T_w}{\frac{W}{AK} + \frac{2W}{CG} + \frac{1}{2}} = \frac{416 - 150}{\frac{200{,}000}{15{,}000 \times 3.6} + \frac{2 \times 200{,}000}{24{,}000 \times 30} + \frac{1}{2}}$$

$$= \frac{416 - 150}{3.70 + .55 + .50} = 56 \text{ deg.}$$

equals rise in temperature of feed water in passing through economizer.

It is then seen that, on the basis of these computations, if the excess air is increased so that the flue gas per pound of coal increases from 21 to 30, the computed temperature rise in the economizer drops from 73 to 56. In other words, the usefulness of the economizer will be reduced to three-quarters of its original value.

It should be borne in mind that the chief use of these formulas is for purposes of comparison and checking. There is ordinarily too much doubt about the exact values of K and G, for highly accurate results to be expected.

In the foregoing formulas no account has been taken of radiation losses. These losses have very little effect on the result, especially if the apparatus is insulated. In other respects the accuracy of the formulas depends chiefly on the accuracy of the substitution valves.

Setting Valves on Allis-Chalmers Diesel

By Paul Keller

Each make of Diesel engine has its individual peculiarities of adjustment, especially as to the valve setting. While an engineer who is at all familiar with one make of Diesel can soon learn how to set the valves of any make, still it is well to be posted on all makes.

Every engineer is familiar with methods of determining the piston dead-center points. In setting the valves of an Allis-Chalmers Diesel engine, any of the usual methods of marking the dead center can be followed. With the dead-center lines marked, the next step is to determine the timing of the exhaust, admission and fuel valves.

The manufacturer will furnish a valve-timing diagram for the particular engine. If this cannot be obtained in time, the diagram shown in Fig. 1 will prove close enough for all practical purposes. The entire valve arrangement appears in Fig. 2, which shows the exhaust and admission valves actuated by a single eccentric. Before going into the timing, the eccentric pins of the rolling cam on the exhaust and admission valves should be set. The line of centers of the eccentric and pin of the rolling cam should be at an angle of 22 deg. with a horizontal line through the pin center. The engine should now be turned over on the power stroke until the exhaust-valve opening point is reached. The point where the valve actually opens can be determined by the resistance of the cam roller when spun by the fingers. The moment the roller touches the valve stem, the roller refuses to spin. This, however, is not entirely accurate. A better method is to chalk the valve bonnet, punch mark the valve-stem nut, and use a pair of dividers somewhat as in Fig. 3. If a line is scratched on the bonnet when the roller is entirely out of contact, this may be used as a base line. Then, as

the flywheel is rotated it is very easy to determine when the valve opens, for the dividers will now make a second line slightly below the first scratch. A point should be marked on the flywheel to denote the valve-opening position and the wheel then turned until the closing point is reached. If the timing checks within one or two degrees with the diagram in Fig. 1, it is not necessary to alter the timing. The lift of the exhaust valve should

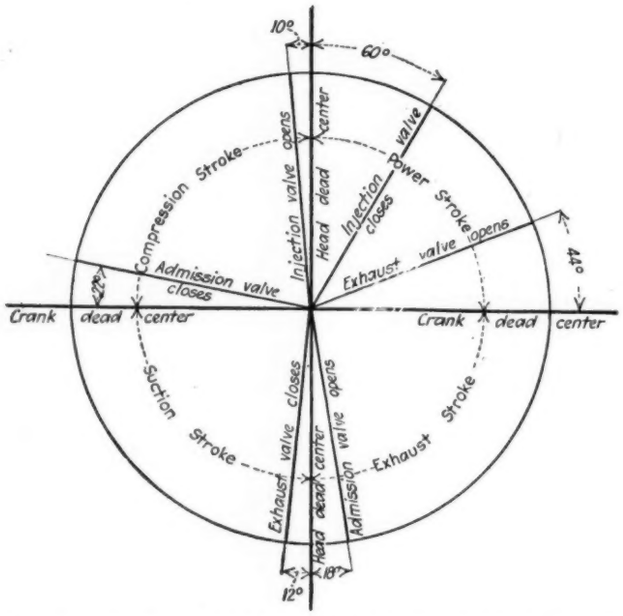


FIG. 1. TIMING DIAGRAM OF ALLIS-CHALMERS DIESEL

be between 1 and $1\frac{3}{8}$ in. The same process is followed with the admission valve.

It will be noticed that on a two- or a four-cylinder engine the timing of the valves of the inside cylinders will not quite check with the diagram. This is due

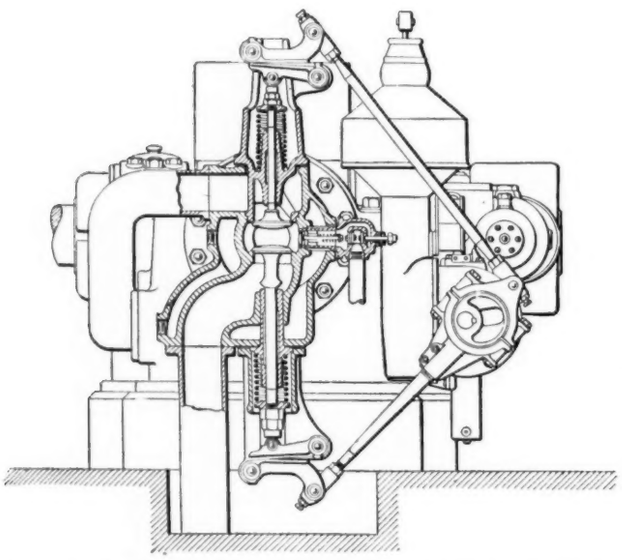


FIG. 2. VALVE ARRANGEMENT ON ALLIS-CHALMERS DIESEL

to the fact that both sets of valves are handled by the one eccentric on the layshaft and the clearances will cause this alteration. The valve lift can be altered by lengthening or shortening the eccentric rods. In a twin-cylinder engine the reach rods can be adjusted for the second cylinder's valves. The lift can also be altered by changing the clearance between the flat and rolling cams or levers; however, this clearance on the exhaust valve should be not less than 0.015 in., while the admission valve clearance can be somewhat less. Any

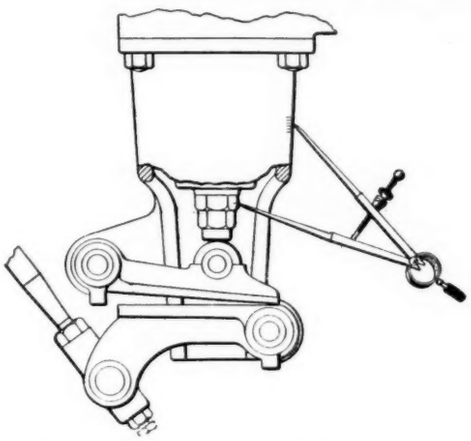


FIG. 3. CHECKING EXHAUST-VALVE OPENING

great amount of clearance will produce considerable pounding. Furthermore, excessive clearance will cause the valve to open late and close early. Also, if the length of the eccentric rod is not correct, the two levers or cams will not come into contact at the proper time, causing early or late opening of the valve. Likewise, shifting the lever-pin angle of eccentricity will change the timing. These parts are but seldom adjusted, and

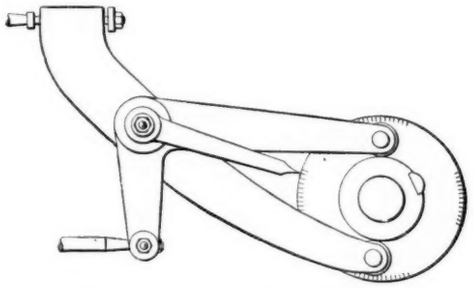


FIG. 4. TIMING SPRAY VALVE

unless an engine has been dismantled, no adjustment of exhaust- and inlet-valve timing is necessary.

The spray- or injection-valve cam is shown in Fig. 4. The timing of the spray valve can be altered by changing the clearance between rocker and spray-valve steam washer or by shifting the cam on the hub; the cam is slotted and held on the hub by two bolts. The cam nose likewise can be shifted to make the valve open early or late.

A handy means of checking the spray valve is shown in Fig. 4. This is a dial placed on the end of the layshaft and held by the screw holding the end cap in place. A pointer fixed to the spray-valve rocker pin allows the dead-center points to be located, as well as the timing points required.

The air starting valve seldom if ever requires any change in the timing. While on the subject it is well to mention that the stuffing-box gland on this valve should never be drawn up hard. If so, the stem will bind, and since the valve is closed by the air pressure acting on the differential-valve areas, the valve may stick open, wasting a large amount of air.

EDITORIALS

Steam Pressures and Superheat

FROM about 1913 to 1918 there was widespread interest in the economic possibilities of boiler pressures up to one thousand pounds with the moderate superheat of one hundred degrees Fahrenheit. At the time there was general agreement that such a pressure and superheat were desirable from a thermodynamic viewpoint, and builders of boilers, turbines and other apparatus involved expressed willingness to assist in the development of equipment to use such pressures and total temperatures.

This was three to eight years ago; what of the practice today? One thousand pounds as a boiler pressure has, as a proposal, been discarded. There are, indeed, few engineers who care to contemplate the troubles that they feel such a pressure will involve. All are interested, but all want the "other fellow" to try it first. Both here and abroad engineers have been content to compromise by using a much lower pressure than one thousand pounds, and a higher superheat than is permissible with such high pressure. The limiting factor in a combination of pressure and superheat is the total temperature, which is around seven hundred degrees Fahrenheit. Above this temperature the materials of construction are likely to give trouble and become unsafe under the stresses imposed by the functioning of the apparatus. American engineers are reluctant to design for seven hundred degrees, for even if the normal total temperature is about six hundred degrees, there will be times when the temperature will exceed seven hundred.

The last year has seen two exceedingly interesting plants designed for high pressure and high total temperature. One is at North Tees station, Newcastle-on-Tyne Electric Supply Company, where the pressure is said to be four hundred and fifty pounds and the total temperature seven hundred degrees Fahrenheit; the other, the "super-power" station in Paris, where the pressure is three hundred and fifty pounds and the total temperature over seven hundred degrees. These temperatures are so crowding the margin of what American engineers have found to be the limit that operation in these stations will be watched with interest.

Experience is proving the correctness of the predictions made in *Power* several years ago that the chief troubles of operating at extremely high pressures and temperatures would not likely lie in the boiler and turbine, but in the valves and piping and expansion joints. These accessories are today the chief source of worry with present high pressures and temperatures. As to steam velocities, several plants are having trouble enough with eighteen thousand feet per minute without wishing to court more of it by the use of smaller piping for far more dense steam. So whatever pressure engineers may attempt, pipe sizes are not likely to change much from those now in common practice. Welding does much to overcome the troubles of gaskets at joints, but engineers have not yet found it expedient to get rid of all joints. It is difficult if not impossible to do so, from the operators' viewpoint.

While the high temperature is serious, it is obvious that what creates the greatest nuisance for the maintenance engineer are the temperature fluctuations that unavoidably occur in daily operation. The equipment using the steam is invariably made up of several different kinds of metals, one part of one material, one of another. The coefficient of expansion is different for each metal, and it is not unusual to have extreme differences in two parts that must preserve tight fit with each other throughout the temperature ranges. The never-ending expansion and contraction tends always toward distortion and permanent set of the parts. The greater the range between initial and final temperatures the more pronounced this action. In a valve, for example, the design and workmanship must be executed with great care so there will not be prohibitive distortion of the seating surfaces. There must be no parting of these surfaces when the valve is closed and later cools off to many degrees below the temperature at which it was closed tight. Wiredrawing across the seating surfaces with such a dense medium as steam at very high pressures soon erodes the surface so badly that they are incapable of holding tight.

Application of High-Head Hydraulic Turbines

THERE has been considerable discussion during the last few years as to which type of hydraulic turbine, reaction or impulse, should be adopted for high-head water-power developments. The use of reaction turbines always permits a higher speed, consequently a considerably cheaper first-cost installation than is the case with the impulse type. Two impulse-type wheels recently installed to operate under a head of a thousand feet and develop thirty thousand horsepower operated at one hundred and seventy-one revolutions per minute, where two vertical-shaft twenty-three thousand horsepower reaction-type units operating under eight hundred feet head operate at six hundred revolutions. These higher speeds for the latter type result in a unit that is much smaller in dimensions for a given output. Since the reaction-type unit is well adapted to vertical-shaft operation, this feature also reduces the foundation and building cost, all of which reduces the initial investment. Moreover it is usually possible to obtain higher efficiencies with the reaction type. With the units referred to, the reaction type developed an efficiency of ninety per cent against eighty-four per cent for the impulse type.

On the other hand, a number of difficulties have been experienced with reaction-type wheels on high-head installations. It has been found that although such units may develop high efficiencies when first put into operation, the efficiency and power both decrease after the unit has been in operation for a comparatively short time, due to increased leakage at the runner seats and

corrosion of the guide and runner vanes. Maintenance charges and repairs also run considerably higher for the reaction type than for the impulse type. As a result of these conditions there has been a growing tendency for the power companies to prefer the impulse-type wheel for high-head installations.

It is the opinion of a number of prominent hydraulic engineers experienced in the application of high-head reaction turbines that the difficulties encountered with reaction wheels can be entirely eliminated by proper care in the design and construction of the various details of the unit. Therefore the leading article in this issue, "Important Features in the Design of High-Head Reaction Turbines," by F. H. Rogers, should be of interest. This article describes a thirty-thousand horsepower unit for the Southern California Edison Company to operate under a head of six hundred and eighty feet. At this time there are a number of high-head reaction turbines being installed or in operation, some of them operating on heads of over eight hundred feet. The results obtained from these units under operating conditions undoubtedly will have a marked influence on the use of reaction turbines in high-head installations in the future.

With regard to the efficiency of the reaction turbine little improvement can be expected over that now obtained with a properly designed wheel. However, before the turbine is an entirely satisfactory unit it must maintain its efficiencies over long periods with reasonable maintenance costs. There are large low-head units that have to their credit thirty months' continuous operation without being shut down. On the other hand, there are many others operating under medium and high heads that have no such enviable record. In the last twenty years hydraulic-turbine designers have made wonderful improvements in the efficiency of these units. In the future there is little doubt that they will make the durability and reliability of the units comparable with their efficiency. In this respect Mr. Rogers' article clearly indicates that the turbine designers and manufacturers are giving careful consideration to these problems.

The Problem of Power and Process Steam

THIS issue of *Power* contains an article entitled "Motor Drive Reduces Power Costs in Cotton-Finishing Plant," by Warren B. Lewis, a consulting engineer, of Providence, Rhode Island. In the particular plant discussed, a certain method of driving equipment and supplying heat to processes had been used for years with the feeling that all was well. When questions came from those of inquiring mind, reasons were given that seemed entirely satisfactory to all concerned. Unfortunately, these reasons were based on assumptions and not on a knowledge of the actual facts. The moment somebody started out to check the facts, the whole plan of operation was seen to be wrong. Further analysis resulted in a new layout that reduced the yearly coal consumed by four thousand tons, or thirty-eight per cent. This does not, of course, mean a net yearly financial saving equal to the value of four thousand tons of coal. Allowance must be made for the overhead charges on the new equipment installed. However, after making all necessary deductions the net gain was decidedly worth while. Such a radical improvement must have produced mixed feelings in the minds of the operating men—delight at the savings and chagrin because such a golden opportunity had remained unnoticed for years before their very eyes.

This story has a lesson for the engineer of every power plant connected with a manufacturing establishment that uses process steam. The lesson is not that individual-motor drive is always the thing. It is not that there is some one particular combination of equipment suitable for all such establishments. The lesson is this: "The fact that things have always been done a certain way does not prove that it is the best way." The man in charge of the production of power and of process heat should study his own plant—the whole manufacturing plant and not merely the power plant—with the utmost care. Too often he has not been able to see the forest because the trees obstruct the view. Let him stand back a bit and consider just what he is trying to accomplish. Then he can return to a closer analysis of the elements of the problem. The manufacturing plant is engaged in processes of production for which it requires certain amounts of power and heat, delivered at certain times and places and under certain operating conditions.

There is usually more than one way of meeting such requirements. It requires imagination and initiative to think of the various plans that may be used. Then it requires practical and technical knowledge combined with hard thinking and good horse sense to pick the plan that will give the best service at the least cost. The engineer that has these qualifications and makes a thorough, systematic study of the power and heat system under his supervision will be repaid either by the savings he will make or by the assurance that he is already operating along the right lines.

Accurate Data Lacking

COMPLETE and accurate records of rainfall and run-off are the only sound foundation upon which to base the development of the water-power resources of a nation. The American engineers who landed in France with the vanguard of the army, to provide an adequate supply of water for the troops, very soon found out that they had left a country exceptionally rich, to land in one exceptionally poor in knowledge of this character. It was found relatively easy to obtain approximate data on the rainfall, but apparently none was available covering run-off. The invariable answer to the query was "about fifty per cent," although in all probability it varied from fifteen to eighty per cent in different parts of the country.

That the French engineers themselves have been hampered by their ignorance is evidenced by the increasing interest accorded to these subjects in French technical magazines. *La Revue Generale de l'Electricite* has already published two articles by A. Contagne, in the issues of August 28, 1920, and May 21, 1921, dealing particularly with the rainfall, and further announces two other articles by the same author on run-off and its relation to stream flow in different regions. These are steps in the right direction, although there is perhaps too marked a tendency to reduce the problem to a series of mathematical formulas of doubtful use to the average engineer. It will be interesting to note whether the author's ingenuity will go so far as to express the run-off by mathematical formulas, which generally prove such handy tools in the hands of European engineers, especially those trained on the Continent.

CORRESPONDENCE

Beware of Snap Judgment

One of the important requirements that a power-plant supervisor should have is the ability to make just and fair judgments. It is an unfortunate fact that many otherwise fine men fall down and make enemies because of their poor method of passing judgment on questions that come before them.

Some time ago I was employed as operating chief engineer in a fair-sized turbine street-railway plant. This plant contained five small turbines and an 18,000-kw. unit, this latter being our main unit, the loss of which would be a very serious matter.

One night my house phone rang and I was informed by my night engineer that our main turbine had gone flat, that the steam pressure had dropped from 200 to 40 lb. in about three minutes, but the auxiliaries were working all right with 200 lb. on the line. In half an hour I was at the plant, and upon my arrival I found the turbine in service, carrying her usual load. My engineer informed me that after telephoning he returned to the turbine room and found that the steam had come up, hence, knowing that I was on my way to the plant, he had put the machine into service again.

Shortly after this our superintendent came into the plant and of course asked for an explanation relative to the cause of our sixteen-minute shutdown. Conditions as told me were repeated, and as I happened to know that we had recently received a good dose of salt water, due to some split tubes in the condensers, I made the suggestion that possibly the steam strainer was plugged with salt.

"Nothing of the kind," said the superintendent. "I know the trouble; the engineer was not onto his job, the turbine tripped out and he was gadding most likely in the boiler room and of course could not catch her and had to shut down and start up again, all of which takes time. My advice is to get a new engineer. Think it over tonight and no doubt you will agree with me in the morning."

We opened the strainer that night and found it as clean as the day it was installed, and inasmuch as the engineer, the oiler and the switchboard man seemed reluctant to give me any information regarding the shutdown, I began to think that perhaps the superintendent was right.

The next morning the superintendent came in about 10 o'clock and asked what I had decided to do in the matter. "My advice," he said, "is that you fire the whole watch, as they have all lied about it and get some help that you can trust." This conversation took place on the turbine platform, and directly over his head was the high-pressure steam gage, piped to the line between throttle and machine. I was facing the gage when the pressure dropped from 204 lb. to about 40 in a few seconds. We had two small units turning over ready for this very happening, and we swung the load onto these machines. While my men were taking care of this matter, I walked back to our gage board and here was the steam back to 200 lb. The superintendent gave one look at the board, one at me and walked out of the station. That was the last we saw of him that day, and no one was fired.

As this was a Sunday, we cut the big turbine out and opened up the throttle and found that salt had collected on the stem inside the valve body. When the valve would close, due to the swing of the load, it would stick and cut off the steam supply. I assumed that the vibration of the turbine as it slowed down loosened up the salt and the valve would then open up and the machine would get full pressure again. After the valve was cleaned and reassembled, we had no further trouble.

Here was a case where snap judgment, if it had gone through, would have worked an injustice on at least two of my men and possibly upon myself.

South Quincy, Mass. A. D. PALMER.

Parallel Operation of Direct-Current Generators

I have read Mr. Kent's and Mr. Bennett's discussion on "Parallel Operation of Direct-Current Generators," July 5th issue, and to my mind the question of equalizer switches depends largely on four things—size of machines, characteristics of machines, load conditions and cost of switching equipment. Small machines may not need as much equalizing as large generators. They are usually on nearly constant loads such as exciter service, or at least on loads that do not change rapidly. Then, too, exciters or isolated lighting generators are generally flat compounded for use with a voltage regulator. These machines may be handled safely with a triple-pole switch, for the change in voltage due to the series field amounts to just about the drop in the armature as the incoming machine takes its share of the load.

On the other hand, large railway generators have very wide swings in load and may be compounded for 10 per cent or more rise in voltage. It would be ticklish work at the best and might be disastrous to parallel the machines unless the equalizer switches were closed first. However, these large machines usually have the negative and equalizer switches on the machine and the

positive switch and circuit breaker on the switchboard. The floor man closes the switches on the incoming machine as it is coming up to speed, and the equalizing current insures that the machine picks up with the right polarity.

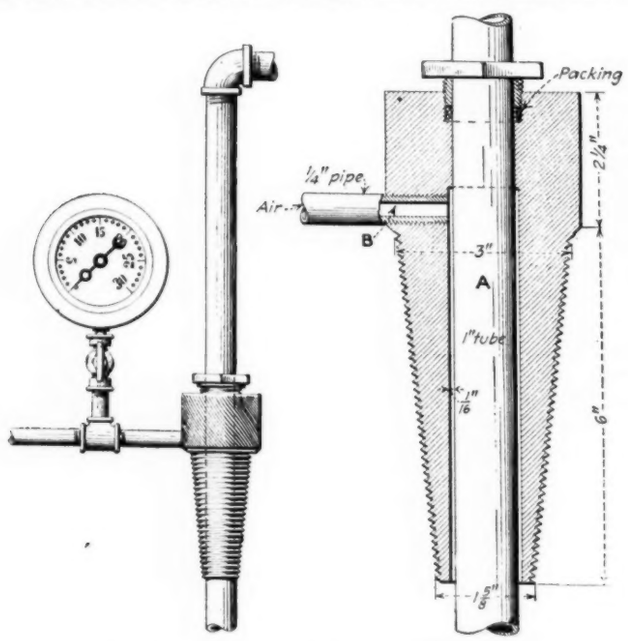
Separate switches for small machines may cost enough more than a triple-pole switch to prohibit their use for exciters or other small machines. The triple-pole switch is usually good enough for this class of service, as the operator readily learns just how much voltage allowance is necessary to get the incoming machine on the bus without a disturbance. Then, too, exciters, or generators, running on building-lighting loads do not get the short-circuits or bumps that are common on railway service and, therefore, are not so liable to reverse their polarity.

If separate switches are desirable, they should be arranged so that the series field of the incoming machines can be put in multiple with the machines already under load, the machine brought up to voltage and then equalized across the open switch. Whether three single-pole switches or a single-pole and a double-pole are used is immaterial so long as the foregoing condition is fulfilled. The fact that the series field always insures the right polarity on the incoming machines was a great help in the old days when large engine-driven generators were common.

N. L. REA, Construction Engineer,
Schenectady, N. Y. General Electric Co.

Emptying Barrels with Compressed Air

A device for emptying liquids quickly from barrels when compressed air is available is shown herewith. It consists of a truncated cone made of hard brass having a thread cut on it and sliding over a discharge pipe. In this particular case the discharge pipe was made of 1-in. brass tube 5 ft. long, one end of which has V-shaped notches sawed out of it as if making a brick drill; an elbow was put on the other end. A hose connection with about six feet of hose completed the pipe.



CONE AND PIPING FOR EMPTYING BARRELS WITH AIR PRESSURE

The brass cone is drilled to take the discharge pipe *A*, allowing a sliding fit. Another hole is drilled and tapped at right angles to the pipe *A* for an air pipe. From the small end of the cone to the point where *B* intersects, the hole in which *A* is fitted is reamed ⅛ in. larger in diameter than the pipe *A*. Packing is placed in a stuffing box at the large end of the cone to prevent air and liquid escaping and still allow the pipe to be moved up and down to accommodate barrels of different size. The small end of the cone must be smaller than the smallest hole on which the cone is to be used and the large end must be larger than the largest hole. The taper should be about one inch in six.

In operation the bung is removed, the pipe placed in the barrel and the cone screwed in by hand. In case it cannot be made up tight enough in this way, a spanner wrench is used in holes provided for the purpose in the knurled surface above the thread. Air is turned on at about 30 lb. pressure and makes its way into the barrel outside the discharge pipe and forces out the liquid. This method was used to such good effect with oil that it was found to be unnecessary to drain the barrels.

Waterbury, Conn. D. A. MCNULTY.

Clearing Installed Electric Conduit by Concussion

The force of concussion has been used in many ways in different tools and appliances, and the most recent tool operated by the explosion of a blank cartridge is a device used to clear obstruction in iron pipe.

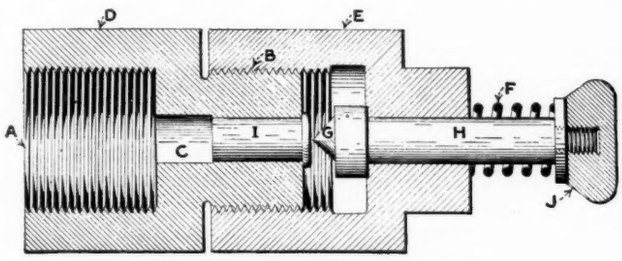
For many years no effective method has been employed to blow out obstructions in pipes, particularly electrical conduits that are placed in the walls or other positions during the construction of a building. Many of these are put in position before the plastering or cementing is completed. It has not been found practical, by experience, to cap the ends of these pipes, although sometimes wooden plugs or waste rags are placed in the exposed ends, and oftentimes these plugs or rags or cement are forced into the tubes to such an extent that when the pipe is sealed into position by the completion of the walls it is practically useless. In many cases conduits had to be abandoned and other pipes set.

A device has been patented, however, that will blow down any foreign matter in conduit such as wooden plugs, waste, cement, plaster, even ice; on a test it blew through 200 ft. of conduit, around elbows, carrying before it all obstructions in the tube.

The principle of the device is concussion, and the tool is simple in construction, easily handled and safe. The electric conduit is provided with screw threads on the exposed ends, to which fittings are screwed. It is on these exposed threaded ends that the device is attached. The illustration shows a fitting in sections, which may be a casting finished in black enamel or galvanized, or it may have its exterior knurled. The interior of one end has screw threads *A*, and the other end is reduced and provided with screw threads *B*; *C* is a small bore through the member *D*; *E* is similar to member *D* and is also provided with threads into which member *D* is screwed. Member *E* is provided with a reduced portion having a longitudinal passage through which the firing pin *H* passes. The firing pin has a firing point *G*. The pin operates freely so that there is no effective pressure until the time

desired. A compression spring *F* on the firing pin causes the pin to rebound after firing the cartridge.

The device is operated as follows: Member *D* is screwed to the pipe, and this member can be made in any size to suit the diameter of the pipe and can be used on two sizes of pipe by means of reducers. A blank cartridge *I* of 32 caliber, is then placed in the bore of member *D* with its flanged end resting on the



SECTION THROUGH FIRING HEAD

face of the passage. Member *E* is then firmly screwed in place. A blow of a hammer on the heads will explode the cartridge, which will cause sufficient pressure to blow out all foreign matter. If a single cartridge does not accomplish the work effectually, a second or even a third cartridge may be used, but as a rule it will be found that one will do the work.

New York City. E. GEORGE CHEUVREUX, JR.

Where the Superheat Went

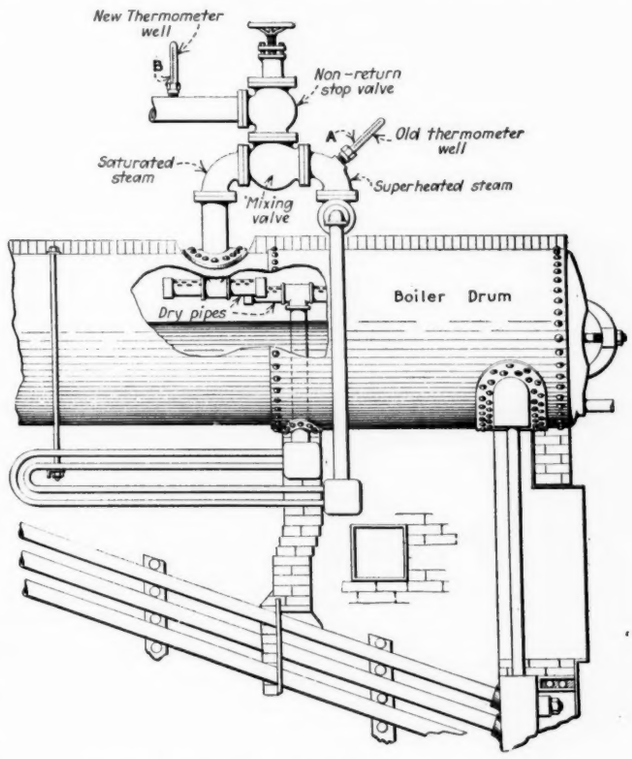
For years, at a certain plant, there had been a standing puzzle and numerous reasons and suggestions had been advanced and various attempts made to solve it. The puzzle was, what became of the superheat? A thermometer placed in the well in the boilers' steam connection showed, from 550 to 600 deg. F., while the thermometer on the pipe to the turbine rarely ever showed above 400 deg. The boilers were operated at 160 lb. gage and had superheaters designed to give 100 deg. superheat. According to the thermometer readings at the boilers, the superheaters were far exceeding their guarantees, but when the steam arrived at the turbine throttle it had lost most of its superheat. Why did it lose it, or how did it lose it?

It was first thought that it was radiation from the steam piping, and although it had a standard covering of 85 per cent magnesia, a second covering the same thickness as the first was applied. Although one could hold a hand indefinitely on the covering the steam temperature at the throttle did not appreciably increase.

The well for the indicating thermometer at the turbine was screwed through the cover of the steam strainer and was three inches long. It was also thought that this thermometer might not be reading correctly, owing to the radiation from the strainer cover, which was not covered, because of frequent removal of the screen for examination. A detachable asbestos cover was put on, but with no added degrees. A deeper well could not be put in because of the peculiar construction of the strainer.

At last it was pointed out that the temperature taken at the boilers at the well *A* was not the real temperature of the steam from the boiler, but that of the steam from the superheater alone. It was further pointed out that the more the mixing valve shut off the superheated steam the hotter it would become, owing to the decrease in flow through the superheater, but the lower the temperature of the combined steam would be. Therefore, to get the greatest amount of superheat, the mixing valve should be set so that all the steam would go through the superheater, although this would mean a lower temperature at the thermometer well in the superheater crossbox.

To prove this, a new thermometer well *B*, as shown, was placed in the pipe to the header. Simultaneous readings were then taken of the two thermometers. With the mixing valve at its original setting the new thermometer gave a reading of 450 deg. and the old one read 595 deg., which showed that the total amount of steam was not up to the required temperature, yet the steam that passed through the superheater was superheated 225 deg. The mixing valve was then moved to the extreme position, causing all the steam to go through the superheater. The temperature of the steam to the header then rose to 510 deg. and the superheater thermometer dropped to 525 deg. As the mixing valve is only a rough casting, it is not steam-tight, therefore, owing to higher pressure in the drum than in the superheater crossbox, because of the pressure drop through the superheater, there was a leakage



POSITION OF OLD AND NEW THERMOMETER

of saturated steam into the outlet, which caused the difference of 15 deg. F. between the two thermometers.

After all the mixing valves were set over to their extreme positions, the steam temperature at the turbine throttle rose to 425 and sometimes to 450 deg., an average increase of about 50 deg. This was gratifying, but still there was something wrong for there was still a temperature drop of over fifty degrees between the boiler and the turbine. With double covering on all the piping this drop seemed impossible.

About this time a recording steam-temperature thermometer was purchased, as it was pointed out that this

recording thermometer was installed and set to agree with the existing thermometer, there would be no gain outside of the recording feature. An 8-in. indicating thermometer was also purchased. The two thermometers were installed in the main 8-in. lead to the turbine. Now for the surprise! These thermometers gave an average temperature of 500 deg. F. As the average of all the temperatures at the boiler outlets was 515 deg., the drop of 15 deg. did not seem excessive. The maximum variation between new and old thermometer was 70 deg. This occurred when the load was light and therefore but little flow through the strainer.

At maximum load the variation was only 35 deg. This change in variation, due to the peculiar position of the thermometer in the strainer, had led to the erroneous impression that the superheat greatly increased with increase of load, which is not always correct. The average superheat is now 125 deg. F., which is above the guarantee.

This is written to show that care should be taken in the placing and installing of thermometers. The thermometer on the boiler was installed correctly and was giving the correct temperature at that place, but its readings were of no value so far as knowing the real temperature of the outgoing steam. It was so placed that all the steam going to the turbine went by it, but it could not measure the temperature of the steam correctly owing to the way it was installed.

The 125 deg. superheat now being received at the turbine throttle is not excessive for a turbine, but it happens that there are two 600-kw. cross-compound engines taking steam from the same header. These engines ran for over six months on this high-temperature steam before the real temperature was ascertained. The cylinders have been examined since and found in good condition, but the potential possibility for trouble was there all the time, unknown to the operators. Fortunately, a high quality of cylinder oil was used, which stood up under the high superheat.

In a modern plant the dependence placed upon the records from recording instruments is very great, therefore extreme care should be taken that these records show real conditions. ALFRED L. POLLARD.

Halifax, N. S., Canada.

The Application of the Producer-Gas Engine

In the March 22 issue of *Power* I noticed an article by J. W. Kitchen, wherein the merits and disadvantages of producer-gas power were discussed. Mr. Kitchen advocates the use of producer-gas engines in combination with steam power. If that refers to the interconnection of steam and gas power stations, it is certainly an ideal solution of the producer-gas engine problem, especially when the gas-power station is located in the neighborhood of a peat bog, lignite field or a number of large sawmills. A producer-gas engine plant should run continuously day and night to avoid the fuel losses that occur when putting a spare producer into service and to reduce the standby losses. Two years ago I had an opportunity to prove this. In a plant I took charge of, it was customary to empty and change one of the three producers every two days. I extended this period to four days and reduced the fuel consumption that month from 22 to 19 tons of anthracite.

When the no-load periods are long, it will pay to install a storage battery in combination with two comparatively small gas engines, one or both running at normal load and the battery charged directly during the low-load hours at a moderate rate, thereby reducing the battery losses and keeping the engines in 16 hours' or even 24 hours' continuous service. The battery will serve to even up the load on the engine.

For instance, a battery consisting of 120 cells, having a three-hour discharge of 1,296 ampere-hours should be charged at a rate as high as 216 amperes, which necessitates a gas-electric unit of 48 kilowatts. However, the cells will gas equally well if charged with 150 amperes, so two engines of 17 kilowatts can be used.

The battery can be floated on the line, charging at low industrial loads, and by discharging at the peak-load periods it serves to enable a small engine plant to carry an industrial load much above the engine rating. Charging during the industrial load, provided no discharge takes place, has to be done with one of the units, in order to avoid excessive gassing of the end cells, located between the charge and discharge brushes of the end-cell switch. However, it will depend on the shape of the load curve whether one or both units should charge the battery. After a heavy discharge, charging will take considerable time and the battery voltage will rise rapidly if there is no industrial load and the whole bank of 120 cells is connected up. Charging with one unit of 17 kilowatts will not be effective on account of the loss of time, consequently two engines are to be put in service. When the main battery starts gassing, one of the units should be shut down in order to apply the finishing rate, which should be half the normal one.

The combination of producer-gas engine and storage battery, if carefully designed and operated, is excellent, as it permits taking advantage of the full-load characteristics of the producer-gas engine. A suitable arrangement of the equipment and the selection of the best combination of units for a particular load will reduce the battery and stand-by losses considerably, thereby improving the load factor.

Klamnaveen, Holland. H. WIELAND LOS.

Discharge Pipe Connections for Centrifugal Pumps

Referring to an article entitled "Discharge Pipe Connections for Centrifugal Pumps," by William A. Frank, page 71 of the July 12 issue, I do not agree with Mr. Frank. If the centrifugal pump referred to in the preceding article, *Power*, June 7 issue, is a good design, no overload can be placed on the driving motor by changing the head or the delivery from normal. Quoting from Centrifugal Pumps," by Daugherty, page 95, as follows:

> It is desirable that the power curve should drop off for the maximum rates of discharge. This makes it possible to install a motor . . . whose maximum capacity is but little more than the power required under normal operation. If, due to a break in the pipe line . . . the head becomes very low, the motor will not then be overloaded. This feature can be attained in the design, either by "throttling" at the eye of the impeller . . . or by making the angle a_2 small.

Angle a_2 refers to the vane angle at exit.

I recently saw a pipe line burst which reduced the head from 440 ft. to 250 ft. and the delivery increased from 12,000,000 gal. daily to over 20,000,000 gal. and there was no increase in power, as shown by the recording wattmeter. On the other hand, a high pressure and low delivery does increase the power input.

Indian Creek, Pa. PHILIP N. EMIGH.

INQUIRIES OF GENERAL INTEREST

Ammonia Expansion Valve—State just what takes place at the expansion valve in a refrigerating plant? R. G. N.

The word expansion valve by no means conveys the true meaning of the functions of this valve. It is a pressure-reducing valve which allows ammonia liquid at the high receiver pressure to flow into the evaporating coils which are at a lower pressure, say, 10-lb. gage. Since ammonia, in order to boil at low temperature must be at a low pressure, and to liquefy in the condenser must be under a high pressure, it is apparent that a valve must be placed between the high- and low-pressure sides to produce the required drop in pressure.

Water Injection in Semi-Diesel Engines—Will water injection in a semi-Diesel engine destroy the lubrication of the cylinder? W. S. M.

If the water is fed in such quantities that it all vaporizes as soon as it enters the cylinder, no lubrication trouble should occur. However, if by the operator's negligence the quantity of water admitted is in excess of the vaporizing ability of the cylinder, the excess might tend to wash the lubricating oil from the piston and cylinder walls. As water in excess will cause the engine to cool off and misfire, giving the engineer ample warning that water fed should be reduced, no great danger exists. On engines having a governor-controlled water valve, excess water cannot enter the cylinder.

Discrepancy Between Meter Readings—Our steam-flow meter shows 96,900 lb. of steam 2 per cent wet used from the boiler. The water meter, however, indicates that we have used 110,500 lb. of water for the same period. Should we not have to use about as many pounds of water fed to the boiler as we use pounds of steam, or may there be so much greater weight of water fed to the boiler than weight of steam used? J. L.

Each pound of steam generated will require a pound of feed water. The excess weight of water shown by the water meter may be due to error of reading, or inaccuracy of the water meter, waste or leakage of water charged as boiler feed water, boiler leakage, leakage and discharge of blowoff water, or use of hot water drawn directly from the boiler. Assuming that the water is accurately metered and that there are no such wastes, losses or uses of boiler water, also that the steam meter is correctly calibrated for the conditions, it may be that there are unmetered uses of steam by auxiliaries, and much escape of steam from the safety valves, either of which would tend to reduce the recorded steam flow to an amount less than the metered weight of boiler-feed water.

Paralleling Compound Generators Before Equalizing.—How may over-compounded direct-current generators be injured by paralleling them before closing the equalizer switch? M. E. A.

When paralleling compound generators, it is essential to close the equalizer and series-field switches before closing the armature switch. Since one machine is usually already carrying the outside load when the incoming machine is thrown on the bus, the load conditions are already instable, all the load being on one machine and none on the other. If the two machines are exactly of the same voltage at the instant of paralleling, the tendency will be for the incoming machine to immediately steal the load and motorize the running machine. They will then blow their breakers and may flash-over at the commutator. A flash-over usually burns the commutator and brush holders severely. The time delay before the flash-over will occur will depend mainly on the size of the units and the difference between the voltages at the instant of paralleling. It is very dangerous, however, to expect that the equalizer switch may be closed in time to prevent damage to the machines.

Setting Valve With Adjustable Riding Cutoff—How is the valve setting done on an engine that is provided with a Meyer adjustable riding cutoff? C. R. Z.

Set the main valve in the same way as an ordinary D slide valve. To set the main valve, first lengthen or shorten the valve rod so the opening edges of the valve travel as far to one side as the other side of the opening edges of the ports. Then set the eccentric around on the shaft so steam is admitted at the beginning of the stroke with desired amount of lead. The lead will usually be enough if it is about $\frac{1}{16}$ in.

After the main valve has been set, turn attention to the cutoff valves. For setting the cutoff valves it must be assumed that with correct setting of the cutoff eccentric and proper length of valve rod, the cutoff valves will be properly spaced on the right and left-hand screw for obtaining the point of cutoff registered by the handwheel. Place the cutoff eccentric about opposite the crank, set the handwheel to register cutoff at half-stroke; turn the engine over in the proper direction of rotation to half-stroke and adjust the cutoff valve rod so the cutoff valve will be just closed. Then turn the engine forward to half-stroke from the other end of the cylinder and observe whether the cutoff valve has just closed. If it has not, take up half of the difference by adjusting the length of the valve rod and the other half by setting the eccentric farther forward or backward around on the shaft. Then turn the engine forward to half-stroke from the end first tried and readjust the valve, making one-half adjustment on the length of cutoff valve rod and the other half by adjusting the eccentric. Then set the handwheel to one quarter cutoff and observe whether the cutoff valve is just closed when the piston is on quarter stroke first from one end of the cylinder and then from the other. If not just closed on the first end tried, bring it half-way by movement of the eccentric on the shaft. Then place the cutoff indicator at half-stroke again and test whether the valve is just closed for half-stroke of the piston first from one end and then from the other end. If not, it must be that the cutoff valves need to be placed in different relative positions on the right and left screw, or the scale of cutoffs may not be correctly laid off when the indicator points to one-half cutoff.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communication and for the inquiries to receive attention.—Editor.]

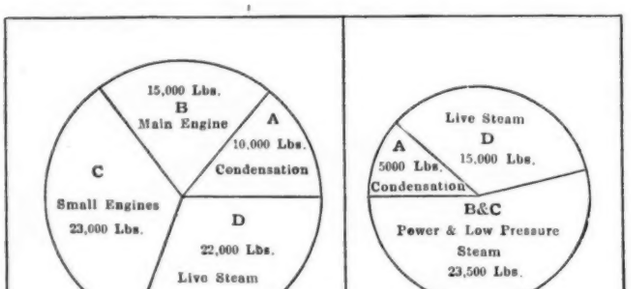
Motor Drive Reduces Power Costs in Cotton-Finishing Plant*

Use of Variable Speed Motors to Replace Steam Engines Lowers Steam Consumption from 70,000 to 43,500 Lb. Per Hour

By WARREN B. LEWIS
Consulting Engineer and President of Providence Engineering Society

THROUGH the use of the variable-speed motor it has been possible to show some very remarkable savings in fuel costs in cotton-finishing plants. This motor has been considered primarily as a variable-speed device which, by reason of its simplicity, perfection of speed control, etc., has outrun all of its competitors. It is not, however, the usually accredited advantages of the adjustable-speed direct-current motor that are paramount in its selection for driving certain classes of machinery used in the cotton-cloth finishing business, but rather the saving of steam that it has made possible.

In the many finishing plants in New England and elsewhere (excluding the few having modernized power plants) are a greater number of small engines that are belted, geared or otherwise connected to machines operating through a considerable range of speed. In addition there is one or more engines exhausting to the atmosphere or running condensing, driving constant-speed machines through the usual lineshafts and countershafts. There are numerous drying machines using steam at pressures of from 5 to 10 lb., individually driven by the small engines, which exhaust into the machines and furnish heat for drying. Small engines



STEAM DISTRIBUTION IN FINISHING PLANT
Left—With individual engine drive. Right—With variable-speed motor drive.

driving machines which, in themselves, do not require heat, will probably be found exhausting into a main connected with drying cylinders, furnishing the steam which these cylinders require in excess of the exhaust of their own engines. Numerous blower systems supply hot air for drying purposes. In many cases the heating coils of these blower systems are supplied with live steam. There are numerous dye tubs, jigs and vats that are periodically filled and emptied. Oftener than not, these are supplied with cold water that has to be heated to the boiling point with live steam.

To the visitor the use of so many small engines is explained by the statement that all the exhaust from these engines is used in drying processes so that nothing would be gained by substituting some drive using less steam per horsepower. The engineering mind, however, will perceive that the actual brake horsepower required for the entire plant does not balance with the heat being discharged from all the engines and conclude that the small engines are justified solely as the means of obtaining varying speeds rather than as a means of producing power as a byproduct from process steam.

The exhaust from these engines is far in excess of that required for drying operations, and the excess becomes available for use in drying cans which condense more steam than is furnished by the driving engines. This leads to the use of a low-pressure "equalizing main" extending into nearly all departments to balance the uses and production of low-pressure steam.

In practically every case a survey of the plant will show that the heat exhausted from the main engines is entirely wasted, except for such as may be used in feed-water heaters. Complete analyses of many of these older plants show that from three to five pounds of coal is used per pound of cloth finished, which indicates at once the importance of the fuel problem.

The equipment of such a plant may be divided into two groups, machines that operate generally at constant speed and those that operate at varying speeds. The consideration of the latter is more important. The installed power required for operating the variable-speed equipment in the assumed plant is as follows:

	Max. Hp. Each
Three singers	6
Two mercerizing equipments	25
Three sets of white cans	10
Two sets of dyehouse cans	10
Six printing machines with their drying cans	30
Five complete tentering equipments	12
One continuous steamer	20
Two ageing boxes	4
One set of starch cans	15
Two open soapers with their drying cans	15
Total	431

Each of these machines will be driven by variable-speed motive power, so that twenty-seven individual steam engines or twenty-seven electric motors will be required with a total capacity of 431 hp. It is, of course, never the case that these machines are all running at maximum speed simultaneously. Extensive experience with this class of machinery has shown an average load factor, as it might be termed, of about 50 per cent.

The constant-speed machines in this plant will require 600 indicated horsepower if they are driven from line shafts. Of this about 50 per cent will be friction losses. If individual motors are used, the constant-speed load will be about 250 kw. This great difference is not due entirely to the difference between mechanical and electrical losses of transmission, but in great part to the fact that the constant-speed machines are operated more or less intermittently.

The boiler plant consists of a number of horizontal return-tubular boilers. The average output is 70,000 lb. of steam per hour, about one-third of which is delivered as live steam to processes where high-pressure steam is used.

The substitution of the adjustable-speed motor makes it possible to eliminate part of the piping system and to reduce the sizes of pipes carrying live steam to processes. If engines are to be operated all over the plant, then boiler pressure, or very nearly that, must be maintained in all parts of the piping system. It is seldom, however, that live steam at a pressure of over 50 lb. is required in the processes. Therefore, with the elimination of the engines the drop in pressure between the boilers and the processes can be made 50 per cent or more. Or, still better, the boilers may be divided into two batteries, one group being operated at 150 lb. or more and delivering steam to the main unit and boiler-feed pump, while the other group may be operated at

*Presented before the Textile Section, American Society Mechanical Engineers, at Providence, R. I.

pressures ranging from 80 lb. down. As a considerable variation of pressure in steam for process purposes is not detrimental, advantage can be taken of the heat stored in the furnaces and in the water within the boilers, so that during periods of heavy demand the fires will not have to be pushed to maintain steam power, while at the slacker periods heat will be stored. This has been done in several plants with the result that the evaporation per pound of coal for the group of boilers furnishing live steam for processes has been very much improved.

The diagram on the left shows the distribution of steam with the original arrangement. It corresponds to a boiler output of 70,000 lb. of steam per hour at 100 lb. gage pressure, the feed water being taken from any source of supply at an average temperature of 100 deg. F. and heated to an average temperature of 170 deg. F.

The area of segment *A* shows the amount of steam condensed in the piping system. Segment *B* is the amount of steam taken by the main engine and *C* is the steam delivered to the small engines. Segment *D* is the steam used as live steam in the processes. Segment *D* presents possibilities for the use of low-pressure steam. In those processes where it is necessary to heat water up to the boiling point, water may be preheated to a temperature of 160 or 170 deg. F. and stored in tanks, from which it may be drawn for use in the dyehouse or in the boiling process, but it is usually far simpler to use directly all the exhaust steam produced.

Conditions After Electrification

If the plant is electrified throughout, the constant-speed motor load will be 250 kw. and the adjustable-speed motor load 210 kw. In this particular plant alternating-current motors (supplied by a 500-kva. generator which may be either turbine- or engine-driven) are assumed for the constant-speed work, while a turbo-generator set to give 230-volt direct current is used for the adjustable-speed work.

The total low-pressure steam uses amount to 23,500 lb. per hour, exclusive of the steam required to heat the buildings. The latter, taken over a twelve-months period, may be ignored as far as its effect on the design of the power plant is concerned.

This plant requires 460 kw. and 23,500 lb. of exhaust steam per hour. A simple non-condensing Corliss releasing-gear engine, direct-connected to a 500-kva. generator, will, between the limits of one-half and full load, develop a kilowatt-hour with 42 lb. of steam per hour at 150 lb. initial pressure and 5 lb. back pressure. A 500-kva. steam turbine will operate at the same steam economy.

To produce 450 kw.-hr. at 42 lb. of steam per kilowatt-hour will require 18,900 lb. of steam, and 90 per cent of this can be delivered to that apparatus that can use low-pressure steam at 5 lb. pressure. This will provide about 17,000 lb. of steam, whereas 23,500 lb. are required. This is a very desirable condition. If a set of drying cans using a large amount of exhaust steam is stopped for a time, then, although the electric power required to drive them represents comparatively little steam, there will still be a deficiency of exhaust steam in the whole system; and it is always possible to introduce live steam.

Excluding that required for heating buildings, the total steam used in this plant, if individual motor drives are used, amounts to 43,500 lb., as shown in the diagram on the right. Of this, 23,500 lb. will produce all the power and low-pressure steam required, 5,000 lb. represents condensation, and 15,000 lb. represents live steam which is used direct. This is a reduction of 26,500 lb. of steam per hour in the amount required to operate the plant. It follows that with an average evaporation of 9 lb. of water per pound of coal over a period of 2,700 working hours, the coal used will be reduced by 4,000 tons a year.

This is by no means the only advantage of the individual motor drive. Finishing plants do not start up at a specified time in the morning and shut down at a specified time in the afternoon, as do many manufacturing plants. As a rule some machines will run overtime, requiring full steam pressure on the entire piping system for the benefit perhaps of two or three small engines. Usually, full steam pressure is maintained on the high-pressure piping system twenty-four hours a day, with the possible exception of Sundays, entailing excessive condensation losses of more than one-third of the steam which can be accounted for as entering into actual production. With the entire elimination of a large part of the high-pressure steam pipes and a reduction in the sizes of those remaining, not only will the condensation losses be decreased, but it will be possible to shut off the live-steam system entirely when steam at that pressure is not required for high-temperature and boiling operations. Boiling operations carried on at night can be served from a comparatively small live-steam main run directly from the boiler house. It is most important to eliminate every foot of pipe that is not absolutely essential and keep steam out of that part of the piping system where it is not needed after the day's work is over.

Again, finishing machines, operated at varying speeds, are subject to many interruptions and are started and stopped or slowed down and speeded up many times daily. Accurate speed control is much simplified by the use of individual variable-speed motor drive.

Commerce Department Establishes Heavy Machinery Industrial Division

The United States Department of Commerce has announced the establishment of a Heavy-Machinery Industrial Division to operate under the direction of Walter H. Rastall, of Dayton, Ohio. The division will take up the problems connected with the marketing of American heavy machinery throughout the world. It is believed that while the growing spirit of nationalism in many countries may hinder the purchase of certain American goods it will improve the situation for American heavy machinery. This is based on the knowledge that these countries must purchase heavy machinery in order to be able to manufacture other products where they desire to be independent of foreign sources of supply. It is believed that this new division will be able to render substantial aid to American manufacturers in finding an outlet for their heavy machinery.

Mr. Rastall is a graduate of Cornell University and a mechanical engineer. He has had much practical experience in marketing machinery, both in this country and abroad, and has just finished an investigation of the markets for American machinery in the Far East. He will keep in close touch with the manufacturers of heavy machinery in this country with the idea of rendering any service possible in connection with the marketing of their products in foreign countries.

The Hetch Hetchy Power Project

Work is progressing rapidly on the Hetch Hetchy tunnels, which are to supply water to San Francisco, Cal. It is expected that the first block of Hetch Hetchy power, amounting to about 66,000 hp., will be available in two and a half years. This power is to be developed in the Mocasin Creek plant. The Pacific Gas and Electric Co. is considering the question of purchasing this entire block of power outright. If the company can be assured of this power, some of its development work can be deferred. If the city desires to bring the power to San Francisco, it must build approximately 150 miles of transmission line. There would also be need of a distribution system if the power were to be used for street lighting and industrial service generally.

A concession for the exploitation of considerable deposits of oil sands and bituminous and carbonaceous shales intercalated with sand deposits in the Canton of Geneva has been taken over by an Anglo-Swiss undertaking. The area of this concession covers a surface of approximately 12,000 acres, and it is being worked principally for the extraction of oil from bituminous sandstone. This stone contains up to 20 gal. of oil to the ton from the surface, and as the works proceed in depth it is expected that the yield will be much greater. A plant is to be erected capable of treating 200 tons of this sandstone per day. It is confidently asserted that petrol will be found, and borings have already begun with this object in view.—*The Engineer.*

Washington News

News of Water-Power Projects and New Applications —Optimism Advised by Comptroller

By Paul Wooton
Washington Correspondent

Based on data submitted by the applicants, the Federal Power Commission has issued the following statements in regard to the project of the Roanoke River Development Co., of Richmond, Va., and the Granite Falls Manufacturing Co., of Hickory, N. C.:

The Roanoke River Development Co., of Richmond, Va., has been granted a preliminary permit for a power project to be located at Buggs Island in the Roanoke River, near Clarksville, Va. The company has two years to investigate thoroughly the proposed development, which is planned to consist of a dam 60 ft. high across the river, and a power house located at the dam, by which means 10,000 primary horsepower can be developed, as well as a secondary development aggregating perhaps 50,000 hp. The primary power will be used for general public-utility and industrial purposes, and the secondary power will be used in operating chemical plants in Richmond. An unusual feature of the project investigation will be study of the effect that impounding waters of the river in its upper reaches will have upon the flow of the stream lower down.

The commission recently issued a preliminary permit to the Granite Falls Manufacturing Co., of Hickory, N. C., for a power development on Wilson Creek, in the northwestern part of the state. The project will serve the City of Lenoir, situated on a branch of the Southern Norfolk & Western R.R., with hydro-electric power for its textile mills and furniture factories. In output of furniture Lenoir is rated third in the United States. The city's industries now depend on steam-generating plants for power, which is very expensive because of the remoteness of the city from coal-mining regions.

There are to be three dams, respectively 34, 40 and 60 ft. in height, each to be connected by a short conduit to a power house. Sufficient water is available to warrant the installation of generating machinery with capacity of 5,000 hp. The company will make accurate surveys of the sites for dam foundations and will compile stream-flow records. It has a year within which to obtain and file data preliminary to a license for the project.

New Water Power Applications

Recently received applications include, among several of lesser interest, the following: Northern States Power Co., Minneapolis; preliminary permit; power project, United States Lock and Dam No. 1 at St. Paul, Mississippi River; 16,000 hp. (all in conflict); public utility. Portland Railway, Light and Power Co., Portland, Ore.; preliminary permit; dam in Clackamas River, tunnel four miles to Oak Grove Creek; public utility. Excelsior Water and Mining Co., Sacramento, Cal.; preliminary permit; three reservoirs, tunnel, conduit, six power houses; 116,800 hp. (39,000 hp. in conflict); public utility.

"Back to Work" Policy Advised

In the course of his address before the National Association of Supervisors of State Banks at Philadelphia, Aug. 5, the Comptroller of the Currency, D. R. Crissinger, among other things said:

There is nowadays entirely too much exploration of industrial and commercial graveyards. Too much attention is being paid to the mistakes of the past and too little constructive thought to the future.

Instead of looking for holes in which to fall, we need to concentrate our thought and energy to ascending the hill just ahead. The inspiration to do, to go, to be off, will bring progress and prosperity. To promote this sentiment, let the resources of the banks be used in starting a new industry and business. You cannot capitalize the mistakes of the past, but stock in the future of our country will yield generous dividends.

Those who have been optimists, those who have played the bull side of the market in America, have always been the winners in this country. Not in two thousand years of well-recorded history has there been a time when any country so securely held leadership and domination. So far as we are privileged now to see, the future is ours. Let us have confidence in ourselves and in one another, and we shall surely realize the utmost that our hopes and aspirations have pictured to us.

Universal Craftsmen Council of Engineers Convention

The Universal Craftsmen Council of Engineers held its nineteenth annual convention at Toronto, Canada, during the week beginning Aug. 1, with headquarters at the King Edward Hotel.

The sessions of the delegates were held in one of the banquet halls of the hotel. The association boasts a membership of 12,000. Besides the representatives of the four councils in Canada there were present delegates from as far west as Seattle, Wash., and as far south as Fort Worth, Tex. This was the first meeting of the organization held in Canada, and a vote of thanks was extended to the efficient local committee and to the members of the Ladies' Auxiliary of Canada for the completeness of the arrangements. One of the important items of business brought before the council was the advisability of extending the membership to all branches of the engineering trade, including electrical operatives. After considerable discussion this subject was referred to the membership committee for final decision. Many prominent engineers were in attendance as delegates and guests.

The Pompeian Room of the King Edward Hotel was tastefully decorated with American flags and Union Jacks and conveniently arranged in booths for the mechanical exhibit. Sixty engineering firms, including British, Canadian and American, occupied the spaces, in the display of power-plant equipment and devices, and attracted the attention of numerous patrons, especially in the evenings.

On Tuesday morning the opening ceremonies of the convention were held, with L. J. Cudbird presiding. After the invocation by Rev. F. E. Powell, Mayor Church welcomed the convention to Toronto. E. M. Carleton made a short address on the Masonic fraternity and R. C. Harris, of the Board of Trade of Toronto, told of the mechanical industries and business enterprises of the city. The necessary committees were then appointed and the convention adjourned to the Pompeian Room for the official opening of the exhibition.

During the week there were several features of entertainment, including auto rides, shopping trips, a dance at Sunnyside Pavilion and a picnic at Island Park.

On Friday morning memorial services were held for members who had passed away during the year.

On Friday evening the convention was brought to a close with a banquet in the main dining hall of the hotel. After the appetizing dinner was served, the following addresses were made: "The Grand Council," H. E. Terry, past worthy chief, and W. G. Warfield, worthy chief; "The Ladies' Grand Council," Mrs. G. Aldridge and Mrs. J. H. Nicholls; "Toronto," T. C. McAree and Russel Nesbitt; "The Exhibitors," R. H. Flett and Edward Flanagan; "The Press," H. Parker and T. H. Fenner. During the evening selections were given by Fred McLean, Charles Leslie, James Reid and Sandy Grant.

At the closing session of the delegates the following grand officers were elected and installed: W. G. Warfield, chief, Newark, N. J.; J. M. Patton, assistant chief, New York; G. Crown, warden, Pittsburgh, Pa.; N. W. Law, guard, Princeton, N. J.; P. H. Early, chaplain, Milwaukee, Wis.; Walter H. Damon, Springfield, Mass., E. H. Bade, Chicago, and L. J. Cudbird, Toronto, board of trustees.

The next convention will be held in Cleveland, Ohio, the first week in August, 1922.

A bulletin issued by the United States Department of Commerce contains the statement that, at a recent electrotechnical exhibition in Germany, Krupp exhibited an electromagnetic process for recovering coal and coke from the residue taken from hard-coal furnaces. The announcement gives no hint as to how this recovery is accomplished.

Popular Ownership of Public Utilities in the Southwest

The Southern California Edison Co. has had considerable success in a campaign to secure wide distribution of its stock among the small consumers of power. On June 24 the company completed the fifth month of its new stock-ownership campaign, at which time it had sold $7,985,000 par value of its common stock. In 1917 the company had a total of 1,864 stockholders, whereas it now has 17,500. Included in this figure are 2,955 employees out of a total permanent payroll of 3,500.

In these days, when there is so much regulation of public utilities, it is evident that wide distribution of stock practically assures a company of the immensely helpful backing of the general public. At the same time it would not be expected that this arrangement would in any way operate against the interests of the non-stockholding public.

Electrical Progress in Italy

Improved labor conditions have enabled considerable progress to be made in hydro-electric construction throughout Italy, and both railways and industrial establishments will soon derive advantage from the provision of a cheaper supply of power. Whereas in 1915 Italy's electric power resources amounted to 2,500,000,000 kw.-hr. per annum, and by the end of 1918 had increased to between 3,600,000,000 and 3,700,000,000 kw.-hr., the provision today is in excess of 4,000,000,000 kw.-hr. The capital of the electric companies has advanced during the same period from 600,000,000 lire to 1,500,000,000 lire ($116,000,000 to $290,000,000 par value). It is estimated that the new capital represented by the additional plants and those arranged for will mean that approximately a further sum of 1,200,000,000 lire ($230,000,000 par) will have been invested. Between 1915 and 1920 new hydro-electric plants totaling 265,000 hp. were completed, while those now under construction will provide a further 400,000 hp. Should expectations be realized in respect of all the projects arranged for, at the end of 1923 there should be a further 1,000,000 hp. available for power and lighting purposes.

Not the least remarkable feature of the electric industrial position in Italy is the almost complete elimination of German interests, notwithstanding the fact that before the war the Teutons practically controlled the market. By far the largest amount of electrical goods was obtained from Germany, while the control exercised through the banks was sufficient largely to exclude both French and British participation. Much has undoubtedly been learned from the Germans, whose highly centralized financial régime enabled them so long and so well to hold the monopoly. It is proposed that hereafter Italy shall produce almost the whole of her own requirements in electrical material, and there is a strong movement toward patronizing home trade only.—*The Engineer.*

Offers which have been made recently to the Engineering Foundation, if supported by financial assistance, will make possible an extensive research in hydraulics. One of the largest hydraulic power companies in America has offered to co-operate with the Engineering Foundation by making its valuable facilities available in any way in which they can be properly used, and an eminent hydraulic engineer has also offered his advice in conducting such a research. To take advantage of this opportunity, the Foundation states that it needs a special fund of not less than $25,000 a year for a number of years. Further information may be obtained from Alfred D. Flinn, secretary of the Foundation, 29 West 39th St., New York City.—*Mechanical Engineering.*

According to a recent bulletin of the Bureau of Standards a thorough investigation is under way in the Metallurgical Division dealing with the micro-changes that occur during the tempering of hardened steels. Some idea of the work involved in this investigation may be gained when it is remembered that it is necessary to take hardness measurements and make microscopic examination of all the specimens of each type of steel for each tempering treatment. Over 800 specimens are involved in the investigation, and the time necessary for polishing is considerable.

New Publications

DIE AUSNUTZUNG DER WASSERKRÄFTE: by E. Mattern, Technische Hochschule, Berlin. Published by Wilhelm Engelmann, Leipzig, Germany. Cloth; 7¼ x 10¾ in.; 1,005 pages; 349 illustrations.

This is the third edition of a treatise on water power. The book is very exhaustive and shows knowledge of the progress of the art in other countries, as well as in Germany.

MAKING ADJUSTMENTS ON THE FULTON-CORLISS VALVE GEAR

Bulletin No. 300, issued by the Fulton Iron Works Co., St. Louis, Mo. This explains in detail and with illustrations the adjustment of the valves of Corliss engines with single or double eccentrics. The directions are very plain and explicit, and the bulletin should be in the library of every steam engineer. It may be had for the asking.

INVESTIGATION OF WARM AIR FURNACES AND HEATING SYSTEMS. University of Illinois Bulletin 120, by A. C. Willard, A. T. Kratz and V. S. Day. Published by the Engineering Experiment Station, University of Illinois, Urbana, Ill. Paper; 6 x 9 in.; 137 pages. Price, 75 cents.

This bulletin is the second report of progress under the present co-operative agreement between the National Warm Air Heating and Ventilating Association and the University of Illinois, for an investigation of warm-air furnaces and furnace heating systems. The tests described were made under conditions corresponding as closely as possible to those of actual operation. The pamphlet is very fully illustrated with photographs and diagrams showing the apparatus used and the results obtained.

PROCEEDINGS OF THE TWELFTH ANNUAL CONVENTION OF THE INTERNATIONAL RAILWAY FUEL ASSOCIATION, Chicago, Ill., May 24-25. Published by the International Railway Fuel Association, Chicago, Ill. Cloth 6 x 9 in.; 410 pages.

This book covers in detail the proceedings of the convention and includes stenographic reports of all the discussions. The following subjects are discussed at considerable length: Sub-bituminous and lignite coal; pulverized fuel; feed-water heaters (both waste-gas and exhaust-steam); oil burning; flat-end grates and ash pans; economic aspect of the fuel-oil situation; use of carbonized bituminous coal; firing practice; fuel saving. All of the foregoing are, of course, discussed entirely from the point of view of locomotive use. However, the discussion brings out many practical points of interest to power plant engineers. The book closes with a list of members of the association.

HIDES AND SKINS AND THE MANUFACTURE OF LEATHER

In this pamphlet of 78 pages, by James Paul Warburg, the process of leather-making is described from the layman's viewpoint. Non-technical language and many illustrations make the explanations easy to understand. Copies will be furnished on application to the Commercial Service Department, First National Bank, Boston, Mass.

Obituary

Major Louis A. Fisher, chief of the division of weights and measures of the Bureau of Standards, died recently at his Washington home at the age of 57. Major Fisher served for a long time in the old Weights and Measures office of the Coast and Geodetic Survey and was put in charge of the Division of Weights and Measures when that branch of the Bureau of Standards was established in 1910.

Personals

R. J. Andrus is now with the Middle West Utilities Co., of Chicago, having left the Central Power Co., Grand Island, Neb.

George A. Ironside is now assistant superintendent of the Hydro-Electric Power Commission, Cameron Falls, Nipigon, Ontario. He was formerly chief operator and maintenance engineer of the Akitiki Power and Paper Co., Iroquois Falls, Ontario.

Business Items

Ford, Bacon & Davis have opened an office at 1421 Chestnut St., Philadelphia.

The Celite Products Co. has announced that E. S. Crosby, formerly sales and advertising manager for the United States and Cuban Allied Works Engineering Corporation, has been made manager of the Eastern District of the Celite Products Co. and will make his headquarters at 11 Broadway, New York City.

Trade Catalogs

The Bury Compressor Co., Erie, Pa., has issued Bulletin 407 describing the Bury, three-cylinder two-stage, variable volume, electrically driven air compressors. Among the novel features of the compressors are the pyramid valves that provide exceptionally large port area and a system of unloading that throws out the two low-pressure cylinders in four steps, one cylinder-end at a time.

FUEL PRICES

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine-run basis, f.o.b. mines):

Coal	Market Quoting	Aug. 2, 1921	Aug. 9, 1921
Pool 1,	New York	$3.15	$3.00@3.25
Pocahontas,	Columbus	3.15	2.75@3.00
Clearfield,	Boston	1.90	1.65@2.15
Somerset,	Boston	1.75	1.50@1.90
Pittsburgh,	Pittsburgh	2.10	2.00@2.15
Kanawha,	Columbus	2.00	2.00@2.25
Hocking,	Columbus	2.15	2.00@2.25
Pittsburgh No.8	Cleveland	2.30	2.25@2.35
Franklin, Ill.,	Chicago	3.15	2.75@3.55
Central, Ill.,	Chicago	2.25	1.90@2.50
Ind. 4th vein,	Chicago	3.10	2.90@3.25
Standard,	St. Louis	1.70	1.75
West Ky.,	Louisville	2.35	1.90@2.55
Big Seam,	Birmingham	2.15	2.00@2.25
S. E. Ky.,	Louisville	2.35	2.15@2.40

FUEL PRICES—

New York—On Aug. 10, Port Arthur light oil, 22@25 deg. Baumé, 4½c. per gal. 30@35 deg., 5½c. per gal. f.o.b. Bayonne, N. J.

Chicago—July 23, for 24@28 deg. Baumé, 40@45c. per bbl.; 32@34 deg. 1¼@1½c per gal. in tank cars f.o.b. Oklahoma refinery, or freight adjusted.

Pittsburgh—On Aug. 8, f.o.b. refinery; Pennsylvania, 36@40 deg., 3¼c. to 3½c. Oklahoma, 24@30 deg. 40c. per bbl., gas oil, 32@34 deg., 1¼@1½c. per gal., 36@38 deg., 1¼@1½c. 38@40 deg. 1½@1¾c.

St. Louis—Aug. 6, prices f.o.b. cars, tank lots; 24@26 deg. Baumé, 35c. per bbl.; 26@28 deg., 35c.; 28@30 deg. 40c.; 32@34 deg. 1¾c. per gal.

Philadelphia—On Aug. 9, 26@28 deg. Baumé, Oklahoma, 50c. per bbl.; 30@34 deg., Oklahoma (group 3) 75c. per bbl.; 16@20 deg. Seaboard, 3¼c. per gal.

Cincinnati—Aug. 8, for 22@28 deg. Baumé, 3c. per gal.

Cleveland—July 29, for 22@28 deg. Baumé, 5c. per gal.

New Construction

PROPOSED WORK

N. J., Camden—The Bd Educ., City Hall, will receive bids until Aug. 22 for motors, switchboards and 35 KW. generator for high school on Blvd.

Pa., Chester—The St. James R. C. Hospital is having revised plans prepared for a 4 story, 108 x 112 ft. hospital. About $250,000. F. F. Durang, 1220 Locust St., Philadelphia, Archt. Noted June 14.

Pa., Philadelphia—The city plans to build 2 new power plants in addition to one now under construction on Fairmount Ave. to furnish power for the Frankford elevated system. About $1,000,000.

Pa., Philadelphia — The Jones Machine Wks., 53d St. and Landowne Ave. is in the market for one 80 hp. and one 125 to 150 hp. oil engines with generators and switchboards (used or new).

Pa., Philadelphia—The trustees of Temple University, Broad and Berks Sts., will soon award the contract for a 6 story, 100 x 150 ft. university at Broad St. and Montgomery Ave. About $300,000. P. H. Johnson, Presser Bldg., Archt.

Pa., Pittsburgh—B. Jansen, Archt., Century Bldg., will soon award the contract for a 7 story, 120 x 250 ft. department store on Stanwis and Duquesne Way, for Joseph Horne Co., Manufacturers Bldg.

D. C., Washington—R. W. Geare, Woodward Bldg., is preparing plans for a 10 story bank, theater and office building. About $2,500,000. Owner's name withheld.

Va., Roanoke—The Superior Anthracite Coal Corp., P. O. Box 106. R. L. Koger, Secy., is in the market for a gasoline motor and 80 to 100 hp. hoisting engine.

Fla., Lake City—Treasury Dept., J. A. Wetmore, Supervising Archt., Wash., D. C., will receive bids until Aug. 30 for 2 semi ambulant quarters, a laundry and a refrigerating plant for the United States Public Health Service Hospital here.

Tenn., Nashville—G. Norton, Archt., Independent Life Bldg., will receive bids until Aug. 29 for an office building on 7th and Union Sts. for The Natl. Life and Accident Co., 302 7th Ave. About $700,000. Heating system will be let by separate contract.

O., Cincinnati — The Hyde Park Dairy Co., 3755 Edwards Rd., is receiving bids for a 2 story, 32 x 85 ft. dairy. About $40,000. Ice machine will be installed. H. W. Voelz, 86 Michigan St., Milwaukee, Wis., Archt.

O., Cleveland—Conrad, Baisch & Kroeble Co., 7318 Wade Park Ave., plans to build a 5 story, 50 x 200 ft. commercial building, including a steam heating system on Euclid and East 82d Sts. About $300,000. A. J. Conrad, Pres. Architect not announced.

O., Cleveland—The County Comrs. plan to build a pumping station and a 1,500,000 gal. reservoir near Wooster Pike and Brook Park Rd. About $375,000. R. F. MacDowell, Court House, Engr.

O., Cleveland—The Ohio State Mortgage Co., c/o Allen Osborn Co., Archts., Rose Bldg., has purchased site and plans to build a 12 story office building on East 9th St. and Huron Rd. About $1,000,000.

Ind., Indianapolis — The city is having plans prepared for a 128 x 192 ft. swimming pool, capacity 610,000 gal., and will require equipment, including motors, pumps, etc. About $60,000. W. Bintz, 1713 South Genesee Drive, Lansing, Mich., Engr.

Mich., Detroit—The Street Ry. Comn., Murphy Bldg., will soon award the contract for a 1 and 2 story car house on St. Jean Ave. About $1,000,000. Separate bids will be taken on heating and power including 3 horizontal cross drum water tube boilers, 250 hp. each, automatic stokers, etc.

Ill., Bloomington—The Illinois Wesleyan College, c/o T. Kemp, Pres., plans to build a college to include library, gymnasium, auditorium, dormitory, school building and steam heating plant. About $700 000. A. L. Pillsburg, Archt. Noted June 28.

Ill., Chicago—Treasury Dept., J. A. Wetmore, Supervising Archt., Wash., D. C., will receive bids until Aug. 25 for a pumping plant and 2 deep wells at the United States Public Health Service Hospital No. 2 here.

Wis., Milwaukee—The School Bd. will soon award the contract for a 1 story, 30 x 44 ft. boiler house and heating plant at Mound St. school. About $50,000. Van Ryn & De Gelleck, Coswell Blk., Archt.

Wis., River Falls—The city, c/o G. Smith, Mayor, will soon award the contract for a pump house and filter basin, centrifugal or triplex pumps, motors and piping. About $20,000.

Wis., Shorewood—The School Bd. rejected bids received July 29 for a high school addition and heating plant on Atkinson St. Bids will be readvertised. Noted July 19.

Ia., Le Mars—W. L. Steele, Archt., 502 U. B. Bldg., Sioux City, will soon award the contract for a 4 story hospital on North Park St. for the Sisters of St. Francis. About $250,000. Noted Aug. 9.

Mo., Fulton—The Fulton State Hospital, c/o M. O. Biggs, Supt., plans to build a power house. About $40,000. Engineer not selected.

Tex., San Antonio—The State Legislative Committee, Austin, has appropriated $100,000 for the construction of a new power plant for the Southwest Texas Insane Asylum. Dr. J. G. Springer, Supt.

Okla., Tulsa—The Silver Plume Lodge, Knights of Pythias, c/o Tulsa Pythian Bldg. Assn., H. L. Sanders, 544 Kennedy Bldg., plans to build a 10 story, 70 x 145 ft. office building on 12th and Main Sts. About $1,000,000. Architect not selected.

Cal., San Diego — The Bureau of Yards and Docks, Navy Dept., Wash., D. C., will soon receive bids for installing four 150 hp. return tubular boilers and 2 vacuum pumps furnished by the Government and furnishing and installing a feed water heater, storage tank, 2 boiler feed pumps, fuel oil pump and heater, 2 steel oil storage tanks, structural steel supports and concrete foundations for all equipment, concrete floor for boiler plant, brick settings for four 150 hp. return tubular boilers, concrete trenches, hall accessories and complete piping systems for all equipment in the boiler plant at the Naval Base Hospital Reservation.

Ont., Goderich—E. H. Darling, Engr., Home Bank Bldg., Hamilton, wants prices on gasoline pumping equipment in connection with proposed chlorination and filtration plant here. About $10,000. Noted June 28.

Ont., Windsor—The Bell Ice Co., 418 Albert St., is in the market for ice manufacturing machinery.

Ont., Windsor—The Essex Provision Co., 1120 McDougall St., will soon award the contract for a 2 story, 125 x 400 ft. cold storage plant, including refrigeration machinery. About $600,000.

Man., Winnipeg — The Winnipeg School Bd. received bids for the installation of a steam heating system in the proposed college which they plan to build on Alverstone and Wellington Sts. from Coller Bros., Fort and St. Marys Sts., $128,658. Bowyer Boag Ltd., 45 Olivia St., $130,880, and Greene & Litster Ltd., 237 Fort St., $131,000. Noted July 12.

CONTRACTS AWARDED

Mass., Everett—The Malden Electric Co., 139 Pleasant St., will build a sub-station at Broadway and Oakland Ave. Contract was awarded to C. H. Tenney & Co., Engineers, 201 Devonshire St., Boston. About $75,000.

N. Y., Iona Island—The Bureau of Yards and Docks, Navy Dept., Wash., D. C., has awarded the contract for improvements to power plant at the naval ammunition depot here to Lord Constr. Co., 105 West 40th St., New York City, $20,939. Noted Aug. 9.

Pa., Philadelphia—The Walnut St. Savings and Trust Co., c/o H. C. Hogens, Archt., 130 South 15th St., has awarded the contract for a 7 story, 94 x 118 ft. office and bank bldg., including a steam heating system at Juniper and Walnut Sts., to the Penn Realty Constr. Co., care of architect, cost plus percentage basis. Noted Aug. 2.

N. C., Greensboro — The North Carolina College for Women has awarded the contract for 3 dormitories and addition to present building to J. A. Jones Constr. Co., Realty Bldg., Charlotte, $364,200. Noted June 21.

Fla., Lakeland — The Federal Ice Co., Blue Island Ave., has awarded the contract for a 2 story, 100 x 350 ft. ice plant to Bloom Suick Co., City Hall Sq. Bldg., Chicago, Ill. About $250,000.

O., Dayton—The Bd. Educ., has awarded the contract for a school on Jackson St., to Drummond & Miller, 4500 Euclid Ave., Cleveland, $347,150. Heating system to W. Haas, 429 East 3d St., $13,674.

O., Dayton — The Sisters of Precious Blood, 305 Xenia Ave., have awarded the contract for a 3 story convent and chapel on Salem Pike to Danis-Hunt Constr. Co., 907 Schwind Bldg., $850,000.

Ill., Alton—The state, c/o Alton State Hospital, Dr. G. A. Zellar, has awarded the contract for a hospital to include six 1 and 2 story hospital buildings to Modern System Constr. Co., Commercial Bldg. About $250,000.

Ill., Dahlgren—The city will build a 1 story addition to electric light and power plant. About $25,000. Noted May 10.

Wis., Sheboygan — The Security Natl. Bank has awarded the contract for a 7 story, 80 x 108 ft. bank and office bldg. to P. Riesen's Sons, 1018 Humbolt Ave., Milwaukee. About $600,000. Noted May 23.

Mo., Kansas City—M. D. Woodling, Reliance Bldg., has awarded the contract for a 10 story, 100 x 124 ft. public garage at 1219 Wyandotte St. to Fogel Constr. Co., Reliance Bldg. About $600,000.

Mo., St. Louis — The Excelsior Laundry Co., c/o G. P. Shaum, Grand and Bell Sts., has awarded the contract for a 2 story, 119 x 127 ft. laundry and power plant on Grand and Washington Sts. to Fruior Colon Constr. Co., Merchants Laclede Bldg. About $200,000. Noted May 3.

Cal., Los Angeles—Klein-Norton Co., 253 South Los Angeles St., has awarded the contract for a 12 story, 69 x 200 ft. loft building on 7th and Wall Sts. to the United States Constr. Co., Hellman Bldg. About $500,000. Steam heating system will be installed.

Ont., London—The city has awarded the contract for electrically driven sewage pumps to the Northern Foundry Co., 88 St. David St., Saulte Ste Marie, $9,378.

Ont., Ottawa — The city, F. H. Plant, Mayor, City Hall, has awarded the contract for a 5 story, hospital on Carling and Parkdale Aves., to A. Garock, Regent Theater Bldg., about $2,500,000. A vacuum steam heating system will be installed.